The
ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

VOL. X. No. 3 MARCH, 1949

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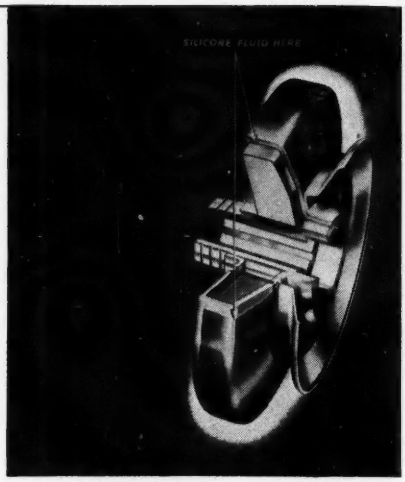

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THE ENGINEERS' DIGEST

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IN THIS ISSUE

IN the last few years, reports have appeared in various Russian journals referring to what was, and still is, claimed to be an electrolytic process of cutting metal. The latest publication of this kind is concerned with the electrolytic sharpening of cutting tools and is reproduced in this issue on page 95. In previous papers, the explanations given of this process were rather obscure, and here again the theory advanced is open to some doubt. In the author's opinion, it is necessary to combine mechanical rubbing action with the electrolytic effect in order to secure the continuity of the latter, and this is achieved by alternately making and breaking the contact. In other words, a principle similar to that embodied in disintegrators used for the removal of broken taps, screws, etc., is employed, relying upon alternately making and breaking an electric arc between an electrode and the piece to be removed and continuously directing a cooling stream of water upon the latter.

Why this should be the case in the Russian machines is difficult to see, particularly if it is borne in mind that no such complications arise in the electrolytic polishing process. Since publishing the article " Electro-Erosion of Hard Metal and Steel " by T. P. Rekshinskaya in our July, 1947, issue, it has come to our knowledge that, for some time past, British engineers have been engaged in the development of cutting processes employing the direct action of electric current. According to British investigations, a cutting phenomenon can be utilized which is of quite a different nature from that described by Russian workers. Investigations and developments in this country have now arrived at the stage where their publication appears to be warranted, and we expect to publish shortly an article by D. W. Rudorff, who has conducted a great deal of research in this field for some years past.

* * *

According to reports which have recently appeared in some British journals, a series of inventions made by German scientists enables the direct conversion of black coal into " hardened " coal, which is regarded in Germany as paving the way for the development of a number of new industries.

Among the products already made of hardened graphited coal are roller bearings and races, which are said to have shown remarkably little wear after months of service in various types of machinery. It is claimed that the saving in lubricant consumption is considerable, since the graphite content makes lubrication unnecessary.

We should have liked to present our readers with full technical information on this patented process, but details are not yet available. However, we have consulted the Board of Trade, German Division, in an effort to discover what was done in this field in Germany in the later stages of the war. We were fortunate to be able to find some original correspondence and research reports taken from the files of the Krupp A.G. in Essen, and from these sources we have extracted as much information as possible and prepared the report on " armour " coal given on page 80.

* * *

The paper on " Tools and Techniques of Management," on page 82 of this issue, has been contributed by the Management Division of the American Society of Mechanical Engineers. It was prepared subsequent to a discussion of the author's lecture delivered during the Management Section session at the Spring Meeting in March, 1948, and may, therefore, be considered as expressing the broad opinion of this important body, although divergencies in detail are not necessarily excluded.

In a preamble the author explains that the subject matter is not new, but he hopes to have provided food for thought so that the result may be of value to the reader in his everyday practice of industrial management. The author goes on to make the open admission that what has been developed is only a hypothesis—and as such may be subject to future modifications—but " Tools and Techniques of Management " as explained in this article are based on practical experience and have emerged from inductive methods ; it can be said that the leading ideas have stood the test for more than twenty years. The limitations in social science which by nature are associated with these ideas are neither regarded as restrictive, nor should they be considered as an excuse for any weakness which may appear in reasoning.

These modest words are not thought to do adequate justice to the article, the value of which can be regarded as outstanding among the innumerable articles published in American literature, for it signposts the road from the viewpoint of the past to that of the present or future without any interruption in such development.

Continued on page 102

Atomization of Liquids by Centrifugal Nozzles

By I. I. Novikov. (From *Journal of Technical Physics*, Vol. 18, No. 3, 1948, pp. 345-354, 3 illustrations.)

The arrangement of a centrifugal nozzle is shown on Fig. 1. The liquid enters the nozzle chamber through the tangential port of small radius r_{BX}. The tangential entry results in a deflection of the liquid in the chamber. Let w_0 denote the inlet velocity, and R the Radius of the nozzle chamber, the moment of momentum of the liquid in the chamber will be w_0R (per kg liquid).

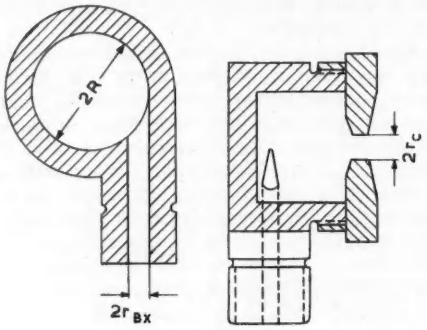


Fig. 1

As the movement inside the chamber is a high velocity phenomenon, the viscosity of the liquid may be neglected; therefore, the law of the conservation of the moment of momentum holds good inside the chamber. Thus

$$wr = w_0R \quad .. \quad .. \quad .. \quad (1)$$

w being the tangential velocity of the liquid at a point at distance r from the nozzle axis.

The axial component of the velocity u, the tangential component w, and the pressure p, must satisfy Bernoulli's equation, viz.

$$p/\gamma + w^2/2g + u^2/2g = H \quad .. \quad (2)$$

The pressure inside the chamber increases at a rate depending on the distance from the axis as follows:

$$dp/dr = \gamma/g \times w^2/r \quad .. \quad .. \quad (3)$$

It is seen that the axial velocity is a constant for all points of a cross-section perpendicular to the axis which are equidistant from the axis.

Eqs. (1), (2) and (3) show that the orbits of the particles of the liquid inside the chamber will be helical. According to hydrodynamic laws this must lead to the formation of an "air vortex" inside the chamber, i.e. a cylindrical cavitation of radius r_B free of liquid particles. This air vortex starts on the back wall of the chamber and issues through the nozzle into the outer air. Its radius r_B is related to the axial velocity by the continuity equation

$$\pi (R^2 - r_B^2)\, u = \pi r_{BX}^2\, w_0 \quad .. \quad (4)$$

and, moreover, has to satisfy the condition $p_B = p_{atm}$, i.e., the hydraulic pressure on the surface of the air vortex must be equal to atmospheric pressure.

Of the equations (1) — (4), only three are independent. However, the number of unknown quantities being 4, the movement inside the nozzle is not yet fully determined. The requisite relation is supplied by the condition (found by G. N. Abramovich) that the delivery per unit time of the nozzle must satisfy

$$\partial G/\partial r_{BC} = 0 \quad .. \quad .. \quad .. \quad (5)$$

where r_{BC} is the radius of the air vortex at the nozzle outlet (or jet).

The system (1) — (5) then yields for G and μ

$$G = \mu \pi r_C^2 \sqrt{2g\,H} \quad .. \quad .. \quad (6)$$

$$\mu = \frac{\alpha \sqrt{1-\alpha}}{\sqrt{1-\alpha+\alpha^2 A^2}} \quad .. \quad .. \quad (7)$$

where r_C is the radius of the nozzle jet, μ the efflux coefficient of the nozzle, and α and A two dimensionless parameters, depending on the geometrical dimensions of the nozzle as follows

$$A = \frac{1-\alpha}{\sqrt{\alpha^3/2}} \quad .. \quad .. \quad .. \quad (8)$$

$$A = \frac{r_C R}{r_{BX}^2} \quad .. \quad .. \quad .. \quad .. \quad (9)$$

$$\alpha = 1 - r_{BC}^2/r_C^2 \quad .. \quad .. \quad .. \quad (9')$$

where α is always positive and < 1, whilst A may take any positive value between 0 and ∞.

The spraying angle, or opening angle of the spray cone, β, may be expressed as follows

$$\tan \beta/2 = w_C/u_C$$

(w_C and u_C being the components of the efflux velocity), Using the above equations, it is easy to show that

$$\tan \beta/2 = (1-\alpha)/\sqrt{\alpha/2} \quad .. \quad .. \quad (10)$$

Eqs. (6) — (10) are due to Abramovich[1]. For comparatively high velocities and pressures, where viscosity may be neglected, eqs. (7), (10) and (9) show that β and r_{BC} are independent of the pressure head H before the nozzle, whereas both μ and β become functions of the pressure at small pressure values. They then decrease together with the pressure, the more marked dependence being shown by β. This is due to the liquid forming a film of conical shape after leaving the nozzle (Fig. 2), the radius of which increases approximately as

$$\Delta\rho = g/\gamma\sigma \times r_C^2/R^2\, w_0^2$$

owing to the drop to atmospheric value of the pressure on the outside of the film. (In the formula, γ is the surface tension of the liquid). The formula is easily derived from eq. (3) by substituting $2\sigma/r_C$ for the difference between the outside and inside pressures. As the initial velocity w_0 at the nozzle entry varies as H, $\Delta\rho$ will be small for high pressures, and angle β will come sufficiently near to the value of eq. (10), whilst at small pressures it will be much below that value. The internal surface of the film makes an angle

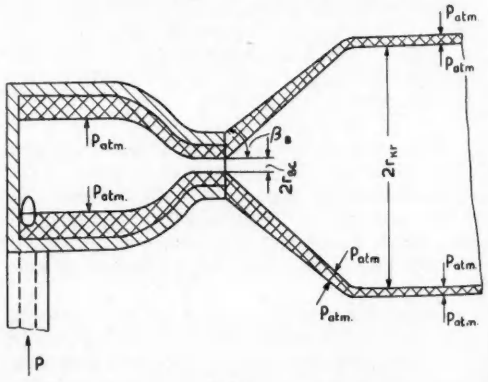


Fig. 2

β_B with the axis which is found from

$$\tan \beta_B = w_{BC}/u_C = \sqrt{2(1-\alpha)/\alpha} \quad .. \quad (11)$$

where w_{BC} is the tangential velocity of the liquid on the surface of the air vortex in the jet of the nozzle. As, according to eq. (1), $w_{BC} > w_C$, $\beta_B > \beta$ and, consequently, the film is continuously thinned as the distance from the nozzle increases.

So far, theory and experiment fully agree on the shape and character of the spray cone. The disintegration of the film under centrifugal forces is counteracted by the forces of surface tension, and so the film retains a margin of cohesive strength and is not broken up directly it leaves the nozzle. The film is possessed of a certain elasticity which maintains the rotation of the liquid outside the nozzle; thus, the law of the conservation of the moment of momentum still applies to the liquid film.

The conical film itself may now be regarded as a tube with elastic walls yielding to the pressure of the liquid. Gradually, the centrifugal forces open up the cone more and more, although, owing to the relation $w^2/\rho = (w_0 R)^2/\rho^3$ they themselves decrease more rapidly with increasing radius ρ than do the forces of surface tension, proportional to σ/ρ. Thus, the increase of the radius ρ is slowed down until it settles to a value corresponding to the equilibrium relation between centrifugal and surface tension forces. The film, therefore, eventually assumes a hollow cylindrical shape of radius ρ and thickness δ. It may happen that full equilibrium between the two kinds of forces cannot be achieved, and this will produce constrictions alternating with dilatations, as has actually been observed in experiments[2], at small liquid pressures. With increasing pressure, the film takes a more nearly conical form, ending in a short cylinder, which soon disintegrates into droplets. This disintegration occurs some distance from the nozzle and not immediately behind it, thus demonstrating the cohesion strength of the film referred to above.

The liquid film has to satisfy Bernoulli's equation, now taking the surface tension forces into account as well; thus

$$p/\gamma + w'^2/2g + u'^2/2g + 2\sigma/\gamma\delta = H \quad .. \quad (2')$$

where δ is the thickness of the film, and w' and u' the tangential and axial components of the velocity respectively.

The liquid pressures on the outer and inner surface of the cylinder are respectively equal to

$$p_{atm} + \sigma/(\rho' + \delta) \text{ and } p_{atm} - \sigma/\rho'$$

where ρ' is the internal radius.

It is now easy to evaluate δ. From eq. (3), and introducing the value of Δp obtaining at small pressure differences between inner and outer surface, i.e., $2\sigma/\rho'$,

$$w'^2/2g = \sigma/\gamma\delta \quad .. \quad .. \quad (12)$$

Re-writing eq. (2') for any point of the internal film surface with the value $w'^2/2g$ introduced,

$$u'^2/2g = H - p_{atm}/\gamma - \sigma/\gamma\,(3/\delta - 1/\rho')$$

As $\rho' \gg \delta$, the term $1/\rho'$ may be neglected against $3/\delta$, thus

$$u'^2/2g = H - p_{atm}/\gamma - 3\,\sigma/\gamma\delta \quad .. \quad (13)$$

Using the continuity equation in the form

$$2\,\pi\,\rho'\,\delta\,u' = \pi\,r_{BX}^2\,w_0$$

and combining with eqs. (12) and (1),

$$\sigma/\gamma\delta = (w_0 R)^2/2g\rho'^2, \text{ or } \rho' = w_0 R\sqrt{\gamma\delta/2g\sigma} \quad (14)$$

whence

$$\delta = \sqrt[3]{g\sigma/2\gamma \times r_{BX}^4/u'^2 R^2} \quad .. \quad (15)$$

or, with eq. (13),

$$\delta = \sqrt[3]{\sigma\, r_{BX}^4/[4\,\gamma(H - p_{atm}/\gamma - 3\sigma/\gamma\delta)\,R^2]} \quad (16)$$

If the liquid enters the nozzle from any reservoir with a pressure head p, and neglecting the pressure losses in the connecting pipe,

$$H = p/\gamma + p_{atm}/\gamma$$

Thus

$$\delta = \sqrt[3]{\sigma\, r_{BX}^4/[4\,(p - 3\sigma/\delta)\,R^2]} \quad .. \quad (17)$$

The thickness of the cylindrical film will, therefore, be smaller, the higher the pressure before the nozzle. In addition, δ depends on the geometrical configuration of the nozzle and on the surface tension of the liquid.

For $\delta > 10^{-3}$ cm and $p > 1$ atm, the term $3\,\sigma/\delta$ in the denominator of eq. (17) may be neglected, i.e.,

$$\delta = \sqrt[3]{\sigma\, r_{BX}^4/4\,p\,R^2} \quad .. \quad .. \quad (17')$$

The hollow cylinder is obviously of a low stability compared with that of a continuous narrow liquid jet, and so is bound to disintegrate into small drops, the total surface area of which cannot be larger than the surface area of the original film. This extreme instability of the cylindrical film explains why it is not being observed in nozzle experiments at all, while the conical part and the transition into the cylinder are being observed. The cylinder disintegrates, directly on being formed, into droplets flying off tangentially and, therefore, not easily observed. The disintegration mechanism sets a very complex problem indeed. However, some of the main features, common to all such phenomena, can be stated. As in the case of a hollow jet disintegrating, capillary waves are propagated along the film, waves of growing amplitude among them. It is these which lead to the eventual disintegration of the film, the initial stage of the process apparently consisting in a separation of the film into single rings, which later on are atomized into droplets. The latter phase is in some ways analogous to the disintegration of a solid jet of liquid. The ring (and drop) dimensions are very likely connected with the original parameters by a resonance condition, as is suggested in the case of a solid jet disintegrating[3], though the calculation of the droplet size in this way is very difficult. Nor are any spectral details of the disintegration obtainable, but a simple formula for the mean diameter of the drops results, and this is seen to agree with experimental data, not only in order of magnitude, but satisfactorily even in numerical value.

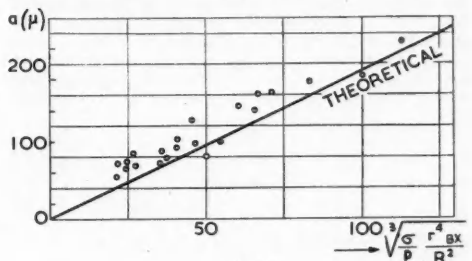


Fig. 3

On the assumption that the total surface area of the drops is equal to the surface area of the cylindrical film, the following obvious expression for the mean diameter (more accurately, the "equivalent diameter" $\Sigma na^3/\Sigma na^2$) of the drops is obtained

$$a = 3\delta \quad .. \quad .. \quad .. \quad (18)$$

where δ, as above, is the thickness of the film in the cylinder. In actual fact, the total area of the droplets will be slightly smaller than that of the film, the numerical coefficient of eq. (18) thus being slightly greater than 3. As experimental data show that the difference between 3 and the actual value is negligible, eq. (18) is proved to be sufficiently accurate. By substitution of δ from eq. (17) or eq. (17') into eq. (18),

$$a = (3/\sqrt[3]{4}) \times \sqrt[3]{\sigma r_{BX}^4/[(p - 9\,\sigma/a)\,R^2]} \quad (19)$$

$$a = (3/\sqrt[3]{4}) \times \sqrt[3]{\sigma r_{BX}^4/p R^2} \quad .. \quad .. \quad (19')$$

The drop diameter is, thus, seen to grow smaller, the higher the pressure head before the nozzle, a decreasing in inverse proportion to the cubic root of p, and varying directly as the cubic root of the surface tension. It is, however, independent of the density of the liquid. Equations (19) and (19′) show that a does not depend on the dimensions of the outlet opening of the nozzle, whereas it depends all the more on the dimensions of the tangential inlet opening and on the radius of the nozzle chamber. The spray becomes finer, the smaller r_{BX} and the larger R. The equation is valid for pressures which are not too small, as at the lowest pressures eq. (19) no longer holds good, viscosity then becoming appreciable. A formula derived by Blinov[3] and based on the Rayleigh theory has the form $\sqrt[3]{r_0^2/p}$, where r_0 represents the exit opening of an idealized jet tube. By the substitution $r_0 = r_{BX}^2/R$ the two results can be shown to be equivalent.

There is not much in the way of experimental data to be found in the voluminous literature on centrifugal nozzles. Nevertheless collating those available, it was possible to plot the diagram in Fig. 3. The heavy straight line gives the theoretical curve $a = \sqrt[3]{\sigma r_{BX}^4/pR^2}$. The experimental points are fairly close to this line. The deviations, especially for large values of the argument, do not exceed the experimental error, while the greater dispersion at small values is due to eq. (19′) being no longer applicable in this range, where it should be replaced by eq. (19) and the numerical coefficient increased from 1·89 to 2·10, in accordance with the theory.

BIBLIOGRAPHY

(1) G. N. ABRAMOVICH, Theory of the centrifugal nozzle (monograph "Industrial Aerodynamics," *ZAGI*, p. 18, 1944).

(2) R. KUEHN, Motorwagen, Nos. 19, 20, 28, 29, 33, 34, 1924; Nos. 2, 4, 1925.

(3) V. I. BLINOV, On the dispersive properties of mechanically sprayed water, *VTI*, 1931.

RUSSIA

The Inclination of Vertical Water Turbine Shafts on Compressible Foundations

By O. V. VYAZEMSKY and V. V. VESELOV.

(From *Gidrotechnicheskoye Stroitelstvo*, No. 9, 1947, pp. 21-23, 3 illustrations.)

WHEN hydro-electric power stations are built, the problem normally encountered is how to deal with the settling of the foundations which may occur after the turbine shaft has been fitted. In some cases turbine shafts have been mounted with a slight incline to the vertical in order to allow for a subsequent expected settling of the foundations, and an example of this is the Svir-3 power station.

The very large Rybinskaya power station had its turbine shafts installed vertically. The weight of the revolving mass of the Rybinskaya turbine is P_1 = 1,100 tonnes, the hydrodynamic reaction P_2 = 960 tonnes.

If the building (and shaft) inclines at an angle i (Fig. 1) the weight P_1 can be divided into a radial component $P_1 i$ and an axial component which is approximately equal to P_1, as the inclination is extremely small.

The radial force is divided between the upper and lower guide bearings, depending on the height of the centre of gravity, and the reactions on these bearings are:—

$$\left.\begin{aligned} P_{upper} &= P_1 i \,\frac{y}{h} \\ P_{lower} &= P_1 i \,\frac{(h - y}{h} \end{aligned}\right\} \quad .. \quad .. \quad (1)$$

Alternators of the umbrella design will have y very near h and P_{lower} will be insignificant. P_{upper} can be assumed to be equal to $P_1 i$.

If i = 0·001, which rarely occurs in practice, $P_{upper} = P_1 i = 1100 \times 0{\cdot}001$ = 1,1 tonnes, which is only a small proportion of the force for which the bearing is designed, and it is obvious, therefore, that an inclination does not impair the normal running of the turbo-alternator.

The requirement for a vertical shaft is justified, however, with a view to the necessity of overhauling the sets. It is essential, when the set is re-assembled, that the axis of the shaft should be perpendicular to the surface of the thrust bearing in order to obtain an even distribution of the load over the thrust pads. If the building has settled on one side, then the specific pressure is unevenly distributed and a bending force will be exerted on the shaft, with consequent deformations. The Rybinskaya Power Station serves as an example. When first checked in 1941, the angle i was found to be 0·00067. The shaft was re-assembled vertically, but a subsequent check in 1943 showed that a further settling of the foundation block had increased the angle by i = 0·0000808 with an azimuth Θ = 44° 08′ to the axis of the sets. For calculation purposes, this angle is considered approximately as i = 0·0001. Referring to Fig. 2, which shows the Michel thrust bearing of the set, 6 and 7 are the inner and outer multi-leaf springs having a rigidity K, with each segment supported by 59·7 × 10⁴ and 65·5 × 10⁴ kg/cm, respectively.

$$K = \frac{192\,Eg}{l^3} \quad .. \quad .. \quad .. \quad (2)$$

The cross-section of a leaf is 18 × 184 mm and the span is 331 and 321 mm. Young's modulus E = 2·1 × 10⁶ kg/cm², the number of leaves is 6, and the moment of inertia is 53·6 cm⁴.

With reference to Fig. 3 it will be seen that when the surface of the thrust bearing inclines by i, the surface of the thrust collar will incline by i_1 because of the difference in compressibility of the springs. The bottom end of the shaft will be inclined by i_2; $i_1 + i_2 = i$. The bending moment around the thrust bearing will be M_0.

If the inner and outer spring diameters are d_1 and d_2 the respective bending moments will be M_1 and M_2,

$$M_1 + M_2 = M$$

$$d\,M = k i_1 z^2\, d\omega \quad = \quad .. \quad (3)$$

where k is a coefficient, and z is the distance of the point from the neutral axis. $d\omega$ is the corresponding element of area

$$M = k i_1 \int_\omega z^2\, d\omega = k i_1 \mathcal{J} \quad .. \quad .. \quad (4)$$

where $\mathcal{J}$ is the moment of inertia.

$$M_0 = M_1 + M_2 = i_1 (k_1 \mathcal{J}_1 + k_2 \mathcal{J}_2) \quad (5)$$

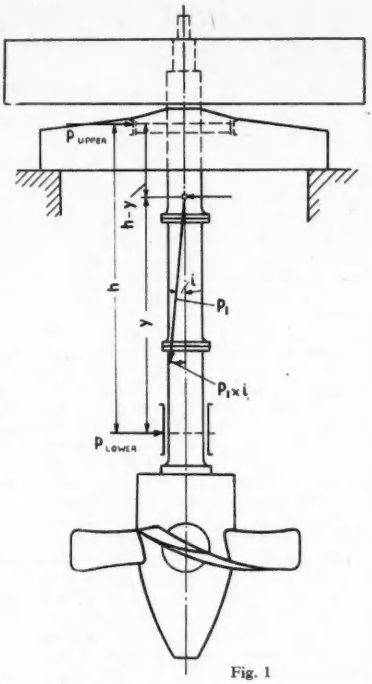


Fig. 1

M_0 is also

$$M_0 = i_2 \frac{3 E \mathcal{J}_{shaft}}{l_{shaft}} \quad .. \quad .. \quad .. \quad (6)$$

From eqs. (5) and (6)

$$i_1 = i \frac{1}{1 + \frac{l_{shaft}}{3 E \mathcal{J}_{shaft}} (k_1 \mathcal{J}_1 + k_2 \mathcal{J}_2)} \quad .. \quad (7)$$

The moment of inertia of the ring-shaped cross-section.

$$\mathcal{J} = \frac{\pi}{64}(D^4 - g^4) = \frac{\pi}{8}(d^3\delta + d\delta^3) \approx \frac{\pi}{8} d^3 \delta \quad (8)$$

where $d = \frac{D + \partial}{2}$ and $\delta = \frac{D - \partial}{2}$

For the practical case of the Rybinskaya set, with $i = 10^{-4}$; $E = 2{\cdot}1 \times 10^6$; $\mathcal{J}_{shaft} = 11{\cdot}6 \times 10^6$ cm^4, eq. (7) would give:—

$k_1 = 18{,}100$ kg/cm^2; $k_2 = 20{,}400$ kg/cm^2

$\mathcal{J}_2 = 2{\cdot}92 \times 10^6$ (*if* $\delta = 1$)

$l_{shaft} = 1064$ cm

From eq. (7)

$i_1 = 0{\cdot}233 \times 10^{-4}$; $i_2 = 0{\cdot}767 \times 10^{-4}$

i_1 introduced in eq. (5) gives

$M_0 = 5{\cdot}25 \times 10^6$ kg cm

The uneven distribution of the specific load

$$\sigma = \frac{P}{w} \pm k_1 i_1 z$$

In the case of Rybinskaya

$\sigma = 1300 \pm 60$ kg/cm^2

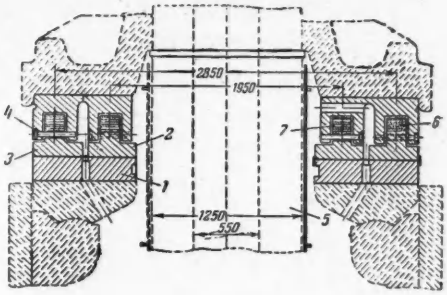


Fig. 2.
(1) Bearing baseplate; (2) inner segments; (3) outer segments; (4) dowel; (5) turbine shaft; (6) outer spring; (7) inner spring.

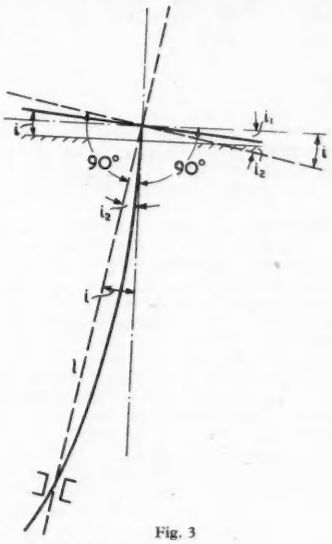


Fig. 3

expressed as a percentage

$$\Delta\sigma = \frac{60 \times 100}{1300} = \pm 4{\cdot}6\%$$

The bending stress in the shaft

$$\sigma_{shaft} = \frac{M_{02}}{\mathcal{J}_{shaft}}, \quad z = 62{\cdot}5 \text{ cm},$$

$$\sigma_{shaft} = 28{\cdot}4 \text{ kg/cm}^2$$

DEDUCTIONS

If the angle of inclination does not exceed $i = 0{\cdot}0001$, the local increase of specific pressure on the bearing surface will be 4·6 per cent, compared with the average specific pressure, and the stress due to the bending moment will be 28 kg/cm^2

It can be stipulated that the maximum permissible increase of specific pressure is in the order of 10 per cent, the permissible inclination being $i_{max} = 2 \times 10^{-4}$ and the stress due to the bending moment $\sigma_{max} = 57$ kg/cm^2. With such inclinations the question of clearance between the turbine blades and turbine casing can become critical and must be given due attention. For that reason, when the maximum permissible inclination is reached, it becomes essential to re-adjust the surface of the thrust bearing.

CANADA

Fatigue Tests of Spot Welds: Improvement of their Endurance Limit by Hydrostatic Pressure

By G. WELTER*. (From *The Welding Journal Research Supplement*, June, 1948, 14 pages, 24 illustrations.)

I. INTRODUCTION

BASED on preliminary test results with spot-welded aluminium alloy sheets, this investigation has been carried out in order to find the best method for strengthening spot welds and the highest possible fatigue load they can carry.

If tested statistically under tension, which produces prevailing shear loads, the spot welds show a satisfactory resistance. However, a different picture of electrically welded joints is obtained when they are placed under repeated loads of a dynamic character, such as found in all mobile structures as motor cars, locomotives, railway and tram carriages, aeroplanes, buses, ships, etc. Under fatigue loads the strength of this type of welds is greatly affected, and the fatigue limit is surprisingly low.

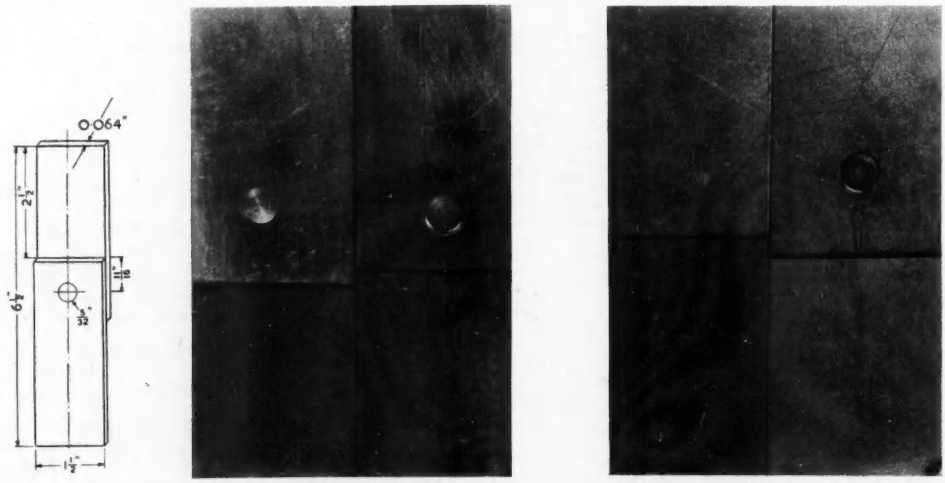
With regard to these extremely low values of the endurance limit of spot-welded aluminium alloy sheets, preliminary tests have been made with the aim of finding some way of improving the fatigue resistance of light alloy sheets. These tests show that under fatigue loads only about 16 per cent of the average static shear strength can be attained in single spots, and double spots seem to support even less than 10 per cent of the average.

The treatment for the improvement of electrically welded joints consists principally in a mechanical process by which extremely high local pressures are applied on both sides of the weld after the welding operation is accomplished and the weld has cooled down to room temperature. It is a general shop practice during electrical resistance welding to apply a certain amount of pressure through the electrodes to the specimen. This welding pressure is necessary to hold the sheets together for an effective flow of the electric current as well as to prevent metal expulsion during the metallurgical welding process in the fused area; for thin sheets this pressure is seldom higher than a few hundred pounds, and only for thicker sheets (0·1 in. and more of high resistance aluminium alloy, for example) does it exceed 1000 lb. It is evident that the welding pressure must be fairly low to avoid deterioration of the welded material which, during the process, is in a melting or semi-melting condition. Furthermore, the usual construction of the welding machines shows that these pressures are not very high, otherwise the arms and frames of this type of machine could not stand these high loads. In the usual welding machines pressure is applied on the electrodes before, during and for some time after completion of the process of inducing fusion of the metal by means of the electric current; it can be changed only within narrow limits if permanent deformation in the welded zones is to be avoided.

On the other hand, the characteristics of the new post-welding treatment to improve spot welds are a *separate* mechanical treatment of the welds *after* the welding procedure is finished as well as applying extremely high local hydrostatic pressures on the spot. This can be done in special machines independently of the welding apparatus or in a combined, special heavily built welding machine after the weld is made and the welded material has cooled to a certain extent.

These extremely high pressures applied over the whole area of the weld are of the hydrostatic type and they exceed the elastic limit of the weld appreciably. By this mechanical high pressure treatment the weld in the fused area of the spot is changed from a cast and annealed crystal structure into a wrought and cold-worked material of much higher mechanical properties, especially those concerning its fatigue resistance. Also the numerous imperfections in all cast metals and alloys as cracks, fissures, porosity, transcrystallization, pinholes, segregation, harmful residual stresses, etc., are seriously reduced in the weld by this process. The mechanically treated homogeneous crystal structure in the spot area is given greatly improved yield strength and a fatigue resistance very much higher than that of untreated welds. To attain this effect it is evident that compression loads higher than those of the usual elastic limit or the yield strength range of the treated materials are necessary. Even much higher compres-

*Ecole Polytechnique, Montreal, Canada.



(a) (a') (b)

Fig. 1. **Some typical specimens of 24ST mild steel and stainless steel before and after mechanical treatment.**

sion loads than the ultimate breaking load of the welded material (at least 3 to 5 times this value) have been applied successfully in producing the desired effect of high endurance resistance in material mechanically treated in this way. Also a few tests were made after the compressed welds had been submitted to a short heat treatment. Other mechanical treatments of the weld, with or without a combination of additional heat treatment have not yet been tried out.

II. RESEARCH PROGRAMME

The following research programme, as proposed by the Welding Research Council (Fatigue of Spot Welds, Sub-Committee of the Resistance Welding Research Committee), has been taken into consideration during 1947:

1. Tests of spot welds from zero to tension in low-carbon steels, Alclad 24ST, and stainless steel 18-8. Perhaps one or two thicknesses could be used, and perhaps one or two spot sizes.

2. The same tests should be repeated with the same materials, the same thicknesses, the same spot size; in short, everything the same except for different varieties of the so-called Amelioration Treatment.

It was further suggested to include section examination of the spot welds and the recording of any helpful supplementary data to secure complete information in regard to the preparation of the specimens, the size and shape of electrodes, amount of pressure, current and timing and similar information on the amelioration treatment, so that all experiments could be duplicated in other laboratories or in practice if found desirable. At the beginning of 1948, the first part of this programme had been carried out and the results obtained during one year of research in the laboratory of Ecole Polytechnique, Montreal, Canada, are described below.

III. MATERIAL AND SPECIMENS TESTED

The three basic structural materials, Alclad 24ST, low carbon steel and stainless steel 18–8, were spot welded in a local aircraft factory on a condenser Type A.-C. Current Welding Machine. To start with, the different alloys were welded under greatly varied conditions in order to determine the optimum working conditions and the best possible physical properties of the three different materials.

Preparation of the Specimens.

1. *Aluminium alloy specimens of* 0·064*-in thickness welded under normal workshop conditions.*—Prior to spot welding all specimens were treated according to the standard practice; special aircraft factory specifications were used in order to remove oxide film, grease and surface impurities. The aluminium alloy samples were degreased, washed and dried and then polished with steel wool and cleaned with a clean cloth; the use of etching solution (hydrofluoric acid, commercial grade), was, after a number of tentative tests, discontinued, as it seemed that the acid-treated specimens did not show much better mechanical results than those simply brushed and cleaned. The machine setting for the 0·064-in Alclad 24ST specimens was as follows: voltage 1450; charge rate 7; 22 condensers of 120 MF each; air pressure 67 psi; the electrodes, file-flattened and paper polished, were of $^{5}/_{16}$-in diam. for specimens Nos. 1 to 12 and $^{1}/_{4}$-in for specimens Nos. 13 to 186. The voltage for these specimens was reduced to 1350 volts. The forging pressure of 1400 to 1450 lb, as indicated by the welding machine, was checked by a Thomson pressure meter; it actually measured 1425 to 1475 lb. The electrodes contour was at the top 4-in radius and flat at the bottom; squeezing time 12 sec, holding time 14 sec. The minimum required shear strength of welded specimens of 0·064-in thickness is, according to aircraft specifications, about 864 lb per spot. Specimens, as used for the fatigue tests, are shown in Fig. 1.

During the last period of this investigation two other series of spot-welded Alclad 24ST alloy specimens (25 samples of 0·064-in and 97 of 0·051-in thickness) were secured through the Welding Research Council, New York. These specimens were also checked by X-ray before delivery, and radiographs were taken of each spot. Prior to spot welding the material was chemically degreased and cleaned. Samples were welded on a condenser energy storage type welding machine. The electrode in the bottom arm had a flat face while top electrode had a 4-in radii domed face.

2. *Mild steel and stainless steel specimens.*—For mild steel and stainless steel specimens the great difficulties encountered in making sound and reliable spot welds in aluminium alloy were practically nonexistent, and after a short time of experimentation fairly sound samples of these two alloys were produced regularly in greater quantities. Nevertheless, every specimen was X-rayed and radiographs taken to obtain the necessary information regarding the soundness of the spots and thus to avoid any unnecessary failure during the long-time fatigue tests with these specimens.

IV. APPARATUS AND TESTING MACHINES USED

The static tests to determine the shear strength of the specimens were made on a Universal 60,000-lb Baldwin Southwark testing machine, using the low range (2400 lb) of this machine. The grips for fixing specimens in the machine were of the flat type. This machine was also used to put the specimens under hydrostatic pressure (see Section V); the applied pressure was checked by autographic records of the machine.

On the other hand the fatigue tests in cyclic tension were carried out on two pulsators, one with a maximum capacity of $\pm$ 5000 lb and about 600 cpm (at the beginning $\sim$ 450 cpm) and another unit of a maximum capacity of $\pm$ 3000 lb and 1750 cpm. Both machines are run by a.-c. motors of $^{3}/_{4}$ and $^{1}/_{2}$ hp. Pulsator No. 1 is shown in Fig. 2 and a close-up of the specimen in Fig. 3.

V. MECHANICAL AMELIORATION TREATMENT OF SPOT-WELDED SPECIMENS

24ST Aluminium Alloy

The mechanical post-welding amelioration treatment of the specimens consisted, as already mentioned, in the application of high hydrostatic local pressure over the welded area of the spot. For this purpose the first information tests were made by placing small hardened steel cylinders of $^{1}/_{2}$-in diameter on both sides of the spot of the welded specimens. The specimen, with cylinders, was placed between the compression tables of the Universal Baldwin testing machine and the highest possible loads were applied to the specimen. The maximum load which could be applied under these conditions without a very marked flow of the material was—for $^{1}/_{2}$-in diam. steel cylinders—about 17,000 to 18,000 lb, that is, about 86,000 to 92,000 psi. This compression load is about twice as high as the yield strength of 24ST alloy. Thus, a fundamental transformation of the structure could be expected, the spot being of the cast or melted type with its low fatigue resistance. After the treatment the cast structure should approach that of the wrought material by showing much better mechanical properties under fatigue stress. This high hydrostatic pressure was repeated several times on different specimens, and the specimens were tested without heat treatment and also following a short heat treatment of the spot material at temperatures of about 400 to 420° F. Somewhat higher temperatures applied during a very short time are also possible. This pressure of 80,000 to 90,000 psi cannot be regarded as the highest possible com-

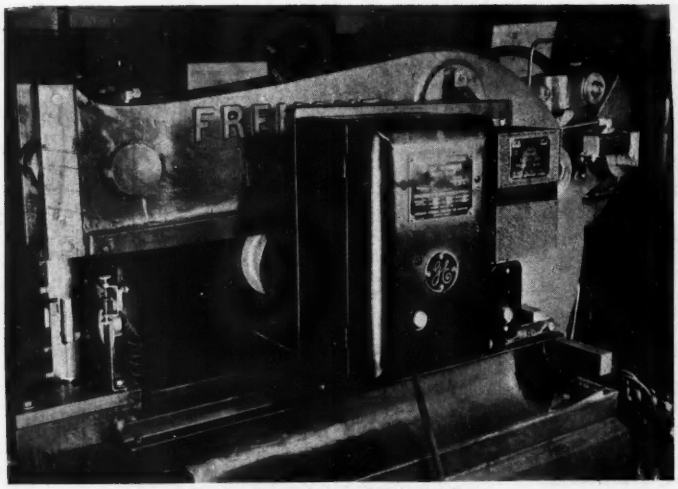


Fig. 2. **General view of Pulsator No. 1.**

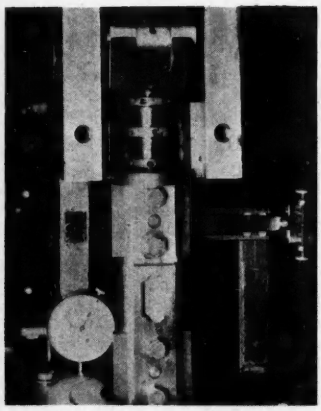


Fig. 3. **Close-up of specimen with grips.**

pression load, as it was not applied hydrostatically from the beginning. Under these local pressures, surpassing the breaking load of the material, the metal already starts flowing visibly. Some cracks under the surface of the steel cylinders could be observed, as well as the beginning of a kind of curling of the sheets, showing a small clear space between the two spot-welded sheets. This clear space is a sign of local flow of the material around the steel cylinders. Furthermore, the shape of the compressing surfaces of the cylinders was varied by using flat faces or surfaces having a certain radius of curvature (radius about 2 to 4 in). This curved compressing face of the steel cylinder was necessary in order to obtain a certain regularity of pressure through the spot, as the concave shape of the spot itself is due to the electrode having a 4-in radius. Actually, the untreated spots showed a thickness of only 0·119 to 0·12-in in the centre of the weld instead of 0·124 to 0·126-in in accordance with the thicknesses of the sheets varying between 0·062 and 0·064-in. A compression of 17,000 to 20,000 lb, with a flat steel piston did not appreciably change the thickness of this weakest point of the spot (0·119-in). Only from 20,000 lb upward was the middle of the spot attained, and under 30,000 lb its thickness was reduced to 0·1125-in. Under 40,000 lb the thickness was only 0·101-in in the middle of the spot. Loads higher than 30,000 to 35,000 lb, representing 155,000 to 180,000 psi, were not applied as the flow of the material under these circumstances was already very pronounced. Under these very high loads the use of a flat compression face of the cylindrical steel piston produced fissures because of the sidewise flow of the material. Furthermore, the diameter of the $^1/_2$-in cylinder was later reduced to $^3/_8$-in, in order to cover and compress only the spot itself and not too much of the adjacent sheet material of the weld.

To be in a position to increase still more the compression stress on the spot, a pressure, in so far as possible of a hydrostatic nature, has to be applied. For this purpose a special device was used, composed of two solid platens fixed together with four strong $^1/_2$-in bolts and having in their middle opposite holes for the guidance of the steel cylinders. These platens, screwed tightly on the welded specimen, prevented the specimen from curling round under the local pressure which could be markedly increased. The higher friction between the platens and the samples prevented the material from flowing sidewise and shielded the compressed section from appreciable exterior changes in its dimensions. In order that the shape of the largely concave spots can be easily acted upon by a uniform pressure over the entire surface, aluminium disks of about $^1/_8$-in thickness were placed between the hardened steel cylinders and the spot weld itself. With the $^3/_8$-in diameter steel cylinders and intermediate aluminium disks, loads of 22,000 lb, that is a unit compression stress of 200,000 psi, could be reached without very marked deformation and plastic flow of the sheet material in the neighbourhood of the spot. At higher loads of about 220,000 psi, under a load of 24,000 lb, a very pronounced flow of the material took place reducing the thickness of the specimen to less than 50 per cent.

By a final device made to prevent the sheets from completely flowing sidewise, somewhat higher pressures, being in this case practically of a fully hydrostatic nature, could be attained. Flat stainless steel sheets were clamped between the platens of the specimen holders so as to prevent the material from flowing in the longitudinal direction near the edge of both sheets. Thus, compression loads of 230,000 psi on Alclad 24ST alloy specimens could be applied without any visible permanent deformation of the welded sheet outside the spot.

Some information tests have been carried out with small and very small specimens (minimum dimensions $^3/_{16} \times {}^3/_4 \times$ 0·64-in) under hydrostatic pressure of over 1,000,000 psi to learn if some fundamental characteristics of light alloys (of the Duralumin type, for instance) can be improved by these extremely high pressures and eventually how an external flow of the material can be prevented.

Ferrous Spot-Welded Specimens.

For these specimens the hydrostatic pressure used on the spot also was as high as possible. The difference lies in the varying diameters of the spots; the compression cylinders had a smaller diameter than those used at first for the aluminium alloy specimens. The first tentative compression load was about 225,000 psi for both alloys, low-carbon steel and stainless steel. No very pronounced deformation to indicate important flow of the material in the area of the spots could be detected. In Fig. 1 (b) are represented some specimens before and after the amelioration hydrostatic compression treatment.

(*To be continued*)

Analysis of Welding Transformer Characteristics

By N. S. SIUNOV. (From *Vestnik Electropromyshlennosti*, No. 1, 1948, pp. 20-22, 6 illustrations.)

IN spite of the fact that welding transformers have been used for many years, there appears to be no simple method of arriving at the transformer characteristics. The author fills the gap by proposing a simple graphical method to that end.

The ohmic resistance of the winding is neglected ($R = 0$) and from the vector diagram the following equation is obtained:

$$I_2^2 x^2 + U_t^2 = U_{2_0}^2$$

from which

$$\frac{I_2^2}{(U_{2_0}/x)^2} + \frac{U_t^2}{U_{2_0}^2} = 1$$

For $\frac{U_{2_0}}{x}$ substitute I_{2k}

$$\frac{I_2^2}{I_{2k}^2} + \frac{U_t^2}{U_{2_0}^2} = 1$$

Where U_{2_0} = no-load voltage.

I_{2k} = short-circuit current.

$$U_t = U_{2_0} \sqrt{1 - \left(\frac{I_2}{I_{2k}}\right)^2} \quad .. \quad (1)$$

Thus, for an analysis of the characteristic, it is only required to know U_{2_0} and I_{2k}. In Fig. 1, the curves calculated in this manner (dotted) are compared with test results, proving that the calculated results are sufficiently close to the actual values for all practical purposes.

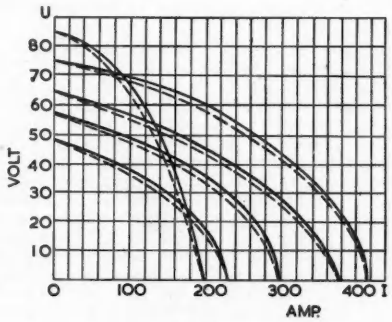


Fig. 1

The formula for the transformer characteristic can be expressed conveniently in specific values.

$$i_2 = \frac{I_2}{I_{2k}} \; ; \; v_t = \frac{U_t}{U_{2_0}}$$

U_{2_0} and I_{2k} are assumed to be unity,

then $$i_2^2 = v_t^2 = 1 \quad .. \quad .. \quad .. \quad (2)$$

From the vector diagram

$$\cos \varphi_2 = \frac{V_t}{V_{2_0}} = v_t$$

Fig. 2 shows a circle diagram for eq. (2) and it becomes evident that the vertical axis is a measure of the power factor.

For the welding current a simple equation applies

$$i_2 = \sqrt{1 - v_t^2} = \sin \varphi_2 \quad .. \quad (3)$$

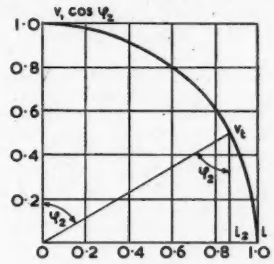


Fig. 2

The output of the welding arc

$$P_t = v_t i_2 = \cos \varphi_2 \sin \varphi_2 \quad .. \quad (4)$$

The condition for maximum useful output is

$$\sin \varphi_2 = \cos \varphi_2 = 0{\cdot}707 \quad .. \quad (5)$$

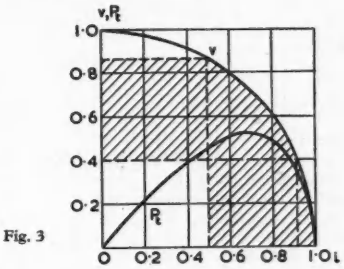


Fig. 3

Fig. 3 shows a graph for the useful output of the arc. It will be seen that when the values for welding current and short-circuit current are close to each other, a higher no-load voltage is required, and also that with higher cosφ the difference between the two currents will increase. The optimum for a welding transformer is within the limits

$$I_2 = (0{\cdot}9 \text{ to } 0{\cdot}5)\, I_{2k} \quad .. \quad .. \quad (6)$$

From this

$$\cos \varphi_2 = 0{\cdot}4 \text{ to } 0{\cdot}87$$

The no-load voltage must, therefore, be within the limits

$$U_{2_0} = \frac{U_t}{0{\cdot}4 \text{ to } 0{\cdot}87} = \frac{25}{0{\cdot}4 \text{ to } 0{\cdot}87} = 62{\cdot}5\text{V to } 29\text{V}$$

It is known that the limit of stability for an arc in an a.c. circuit is determined by the requirement that the no-load voltage should be 1·31 times the arc voltage.

Efforts to lower the no-load voltage have so far been unsuccessful because of the requirement in the past of designing transformers for a wide range of currents.

It is suggested that this requirement should in future be abandoned and the current range limited, to enable designers to reduce the no-load voltage.

Welding transformers having the following standard ratings are proposed:—

50-150 A, 150-300 A, 300-600 A, 600-1200 A, and 2000-4000 A.

This current range limitation will immediately permit of reducing no-load voltage for large transformers to 40 to 45 V. The present usual value of 65 V will only apply to transformers of 50 to 150 A.

Armour Coal, a German Attempt to Utilize Powdered Coal as Material for the Production of Sintered Compacts

(A survey based on original files of *Krupp A.G., Essen*, made available by courtesy of the *Board of Trade, German Division*, Technical Information and Documents Unit.)

FROM German reports which came into the possession of Allied Observers after the German collapse, it appears that from 1943 or 1944 onwards serious attempts were being made to apply powder-metallurgical methods to pulverised coal, thus creating a new material called "sintered coal" or "armoured coal," which possessed unusual properties.

One particular report refers to the experimental pressing and sintering of very finely ground mixtures of bituminous and semi-bituminous coals, and it is stated that such mixtures are highly suitable for the production of sintered products. The first part of the report gives data concerning the compacting and sintering of the following coals:—

Number	Provenance	Ash %	Volatiles %	Caking Index
I	Girondelle II ..	5·32	13·88	51·3
II	Girondelle-Constantin ..	—	17·1	235
III	Briquette-Fines Harbede ..	15·0	9·7	0
IV	Seam Röttgersbank ..	4·46	21·56	—
V	Seam Dickebank	7·08	19·02	180
VI	Geitling ..	4·34	14·62	0

Nos. I and III are non-caking semi-bituminous coals, while No. II is a caking coal. Cylindrical as well as square compacts were prepared from each of the individual coals mentioned above and from mixtures of them. After pressing, the compacts were sintered at a constant temperature of 1000° C for one hour, air being excluded. The sintered pieces were then examined for hardness, density, fracture strength, and specific electrical resistance.

Properties of Compacts made of Coal No. I.

A number of cylindrical compacts were made with compacting pressures of 360, 880 and 1320 kg/cm² respectively, while square compacts were made with pressures of 100,230 and 350 kg/cm² pressure. The relationship existing between compacting pressure and percentage of theoretical density is based upon the true density of the pulverised product of 1·41 grams/cm³. In making the specimen compacts, a constant mould filling of 17 grams for the cylindrical moulds and 20 grams for the square moulds was employed. The curve referring to cylindrical compacts in Fig. 1 shows that densities up to 80 per cent of the theoretical figure could be obtained, while for square compacts a corresponding maximum figure of 70 per cent was registered. This coal was found to have satisfactory sintering properties, the specimens presenting a smooth surface and showing no distortion. On the average, the density of the compact increased by 19 per cent during sintering, while the shrinkage averaged 29 per cent by volume, Fig. 2. The influence of the compacting pressure upon the specific electrical resistance of the sintered specimen is evidenced by the curve given in Fig. 3. This graph also shows the corresponding specific resistance values for commercial Siemens high-resistance and low-resistance materials respectively, which values are seen to lie far below those for the No. I coal. Stress-to-fracture properties as influenced by the compacting pressure of No. I coal are charted in Fig. 3a. It is stated that, in contrast with Siemens-Plania material, a strength of 150 kg/cm² was just attained but could not be exceeded. The hardness figures found for Coal No. I, however, were found considerably to exceed those given for the Siemens high-resistance carbon material, Fig. 4.

Properties of Compacts Composed of Mixtures of Nos. I and III Coals.

Coal No. III proved to be inert owing to its non-caking properties. As will be seen from Fig. 5, maximum hardness values were reached when the percentage of No. II coal lay between zero and four per cent. However, for mixture-percentages lying within the shaded area shown in graph Fig. 5, for some unknown reason good quality sintered specimens could not be obtained owing to spalling. The relationship between strength-to-fracture and mixture percentage for different compacting pressures is charted in Fig. 6, from which it will be noted that the strength decreases with rising percentages of No. III material in the mixture. The functionality between mixture composition and specific electrical resistance of the sintered material is shown in Fig. 7 for compacting pressures of 360, 880, and

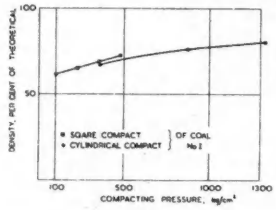


Fig. 1

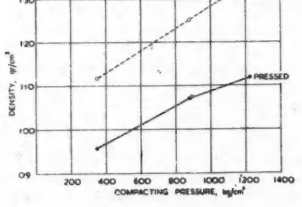


Fig. 2

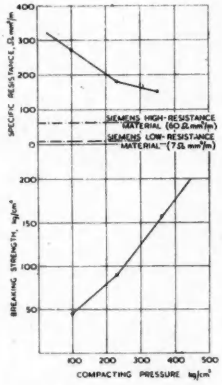


Fig. 3

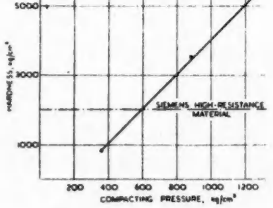


Fig. 4

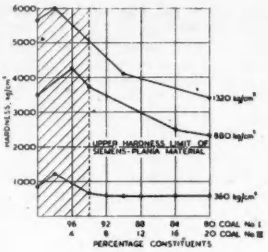


Fig. 5

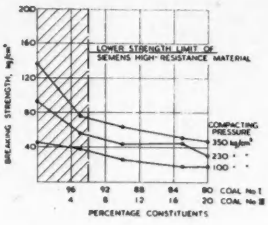


Fig. 6

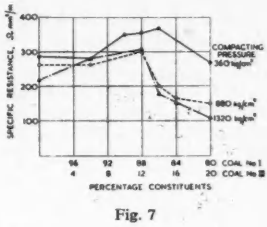


Fig. 7

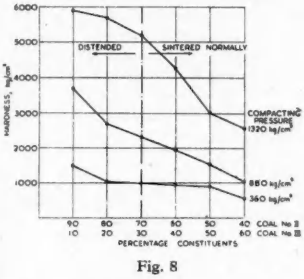


Fig. 8

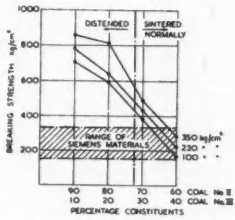


Fig. 9

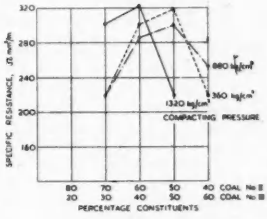


Fig. 10

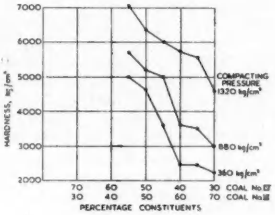


Fig. 11

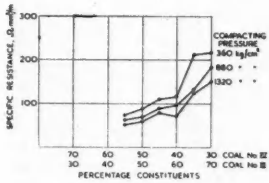


Fig. 12

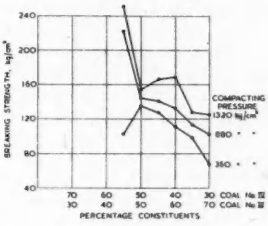


Fig. 13

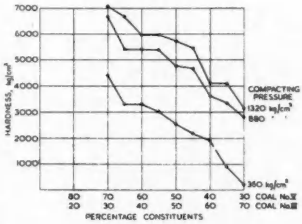


Fig. 14

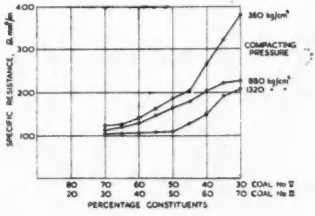


Fig. 15

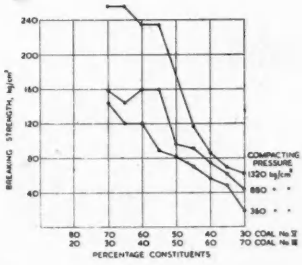


Fig. 16

1320 kg/cm² respectively. It will be noted that with rising percentage of No. III coal in the mixture, the specific resistance at first increases and then decreases, this characteristic applying to all three compacting pressures employed. Both the maximum and minimum values of 358 Ohms and 105 Ohms respectively for the specific resistance recorded lie far above the upper limit of 50 Ohms given for the Siemens material.

Properties of Compacts Composed of Mixtures of No. II and No. III Coals

The hardness values found for mixtures of Nos. II and III coals are charted in Fig. 8, which shows that, for all compacting pressures, there is a steady decrease in hardness with increasing percentages of No. II coal in the mixture. Maximum hardness values are seen to range from 5200 to 1000 kg/cm² according to compacting pressure for a 70 per cent No. II—30 per cent No. III mixture. The higher hardness values obtained with mixtures containing more than 70 per cent No. II coal must be considered meaningless because of the distended and therefore useless product obtained. Stress-to-fracture curves are shown in Fig. 9, which also shows the much smaller figures obtainable with Siemens materials. The considerable influence exerted by the percentage composition of the mixture upon the specific electrical resistance is evidenced by Fig. 10, where the resistance is seen to range between extreme values of 321 and 219, according to mixture composition.

Properties of Compacts Composed of Mixtures of Nos. III and IV Coals.

Referring to the graph Fig. 11, it will be seen that with a compacting pressure of 1320 kg/cm², the record hardness of 7050 kg/cm² was reached with the use of a mixture containing 55 per cent of No. IV coal and 45 per cent of No. III coal. It will also be seen that the

hardness decreases sharply with increasing percentages of the semi-bituminous constituent, but even so, the hardness still remains above that obtainable with other coals. The corresponding curves for the specific electrical resistance are charted in Fig. 12, where it will be seen that increasing percentage of No. III coal leads to a sharp rise in the specific resistance of the mixture. The breaking strength data given in Fig. 13 reveal a record figure of 250 kg/cm² for a mixture composed of 55 per cent No. IV coal and 45 per cent of No. III coal.

Properties of Compacts Composed of Mixtures of No. III and No. VI Coals.

It was found that No. VI coal is entirely unsuitable as it is cannot be sintered satisfactorily.

Properties of Compacts Composed of Mixtures of No. III and No. V Coals.

The hardness values obtained for various mixtures are charted in Fig. 14. Maximum hardness was found to be 7050 kg/cm² with a mixture composed of 70 per cent No. V coal and 30 per cent No. III coal, the compacting pressure being 1320 kg/cm². In Fig. 15, it is shown that, with varying mixture compositions and compacting pressures, a range in specific electrical resistance from 100 to 380 Ohms was covered. Strength-to-fracture varied from 255 to 18 kg/cm² according to composition and compacting pressure, Fig. 16.

Additional Data.

With regard to the fineness of the pulverised material used, one report mentions the following particle-size distribution of a specific sample, viz. :—

Larger than	42 microns	: 4 per cent.
„ „	16 „	: 16 per cent.
„ „	20 „	: 26·5 per cent.
Smaller than	20 „	: 53·5 per cent.

Additions of 17·5 per cent of glue to the pulverised material are claimed to facilitate compacting, and in one case it is reported that such an addition increased the hardness of the final product from 2200 kg/cm² with a compacting pressure of 600 kg/cm² to 6600 kg/cm² when compacting at 800 kg/cm² pressure. Additions of tar or of cumarone resin yield extremely hard sintered compacts, but introduce a tendency to swelling. However, it is suggested that this undesirable tendency may possibly be eliminated by sintering compacts containing tar additions in a carbon-monoxide atmosphere under pressure. The employment of hydrogen atmospheres must generally be ruled out, as this leads to spongy and soft sintered compacts.

Machinability of Sintered Coal Compacts.

Ordinarily sintered coal compacts are too brittle to permit of machining them with cutting tools. This difficulty, however, can be overcome by treatment of the initial material with solutions of alcohol and camphor, or benzene and camphor. It was found, however, that the aforementioned mixtures of No. I and No. III coals with hardnesses of the sintered compact ranging from 400 to 6000 kg/cm² could be machined without any such special treatment.

Uses of Sintered Coal Compacts.

It appears that Krupp in Essen intended the use of sintered coal compacts for sintering crucibles employed for the sintering of hard metal carbides. The standard materials formerly used for this purpose were two qualities of Siemens-Plania graphite—one of high specific electrical resistance, i.e., 20-100 Ohms/mm²/m and of poor machinability, and the other of low re-resistance i.e., 6 Ohms, but of good machinability. These materials were to be replaced by sintered coal combining high specific resistance with good machinability. Sintered carbon compacts were made in the shape of hollow cylinders for insertion as protective bushes in standard graphite moulds. Mention is also made of the use of sintered carbon compacts as bearing bushes.

U. S. A.

Tools and Techniques of Management

By R. W. Elsasser. (From *Mechanical Engineering*, Vol. 70, No. 12, December, 1948, pp. 979-982.)

There is an orderly and relatively precise method available in all management or administrative thinking. An understanding of the method rests upon definitions of terminology which are sharper than those ordinarily experienced. It rests also upon the acceptance and use of a few simple tools and techniques which facilitate analysis for decision and action.

The engineer's anxiety over changing the form or location of matter, over timing the transfer of energy, or reducing the waste of natural forces is usually tempered by his confidence in mathematical prediction. It is a confidence that is born of established principles and familiarity with limitations in their application. The art of application of scientific principle is above all things the art of detecting variations in actual conditions from laboratory conditions ; it is the art of recognizing the departure of reality from the rigid assumptions or standards which by definition are so essential to the validity of the principle.

Should there be any wonder then that the trained technician who expects to find matter predictable is perplexed and impatient over the language of the social sciences? It may provide the engineer with some comfort to know that very few social scientists themselves are able to use the principles in the discussion of which they are so expert. This is no criticism of their inability. The explanation is that they have not given the application of social principle significant attention nor do most of them have any interest in application. They prefer to confine their thinking to those areas of abstraction that are " unsullied by practical application."

It is not surprising, therefore, to discover through experience that in the management of human affairs our language is inadequate, and that the first prerequisite of those who assume the responsibility of directing human affairs is the promotion of common understanding through the development and use of precise terminology. In our relationships with each other we may use the same words, but the chances are almost certain that significant words carry different meanings, or at best they are obtuse and tend to confuse rather than clarify. Every individual who seeks to increase his managerial capacities should become a student of semantics, if only for the proverbial fifteen minutes a day.

The weaknesses in language should never be forgotten in all administrative thinking until we produce a language of management that conveys precise concepts. It will always be the responsibility of management to produce organization unity in spite of the inadequacies of our communicating tool.

Despite the distinction between " management "

and "administration" which is frequently made in some circles, the two terms are used synonymously for our present purposes.

It may be helpful to make a specific distinction between the "academic" and the "practical." In the former case we compare and relate facts, qualitative as well as quantitative, for their own sakes, in our "search for the truth." In practical affairs, though we may not take the time to define it, we have a different aim. The desire to attain or realize an objective prefaces the definition of a standard, and it must precede the recognition of problems in the pursuit of such aims.

The use of remedies rests on a knowledge of causes, of principles established or assumed. If they have been established, we are using the "tools" of science. If they have not been established, we accept hypotheses and then we experiment in action, because act we must. Otherwise we may be "speculating" or "theorizing."

BROAD SCOPE OF MANAGEMENT TECHNIQUES

The subject matter of this paper is presented without limitation in the areas of application. These tools and techniques have proved useful not only in the management of administration of a business unit, but also in the management of non-profit educational institutions, hospitals, public projects, associations, and political groups. They have been proved useful in the management of the affairs of individuals.

It will become apparent that they are, in fact, the techniques of the art of engineering in human affairs, to focus attention upon and to provide standards for comparison, judgment, decision, and action.

Management is the direction of human effort in the pursuit of a specific purpose. The direction of an organization rests upon the sharp definition of that purpose and upon its communication to all individuals within the organization who are charged with some supervisory responsibility. Since management derives its authority ultimately from the individual or group endowed with the power of appointment, it is to be presumed that its purpose at all times must be acceptable to its source of authority. The executive committee of the board of directors, and the board itself are usually a significant part of management with ultimate authority resting in the stockholders collectively, under stipulations prescribed by law. Since those individuals who hold the residual equities in the assets of the business unit enjoy alone the privilege of selecting the personnel who are charged with management responsibilities, it is normal to find top management chosen for its sensitivity to the major aims of stockholders.

At the risk of diversion, let it be acknowledged that some conscientious students of administrative reform have insisted that employees and other groups with a direct interest in the affairs of a corporation should participate with stockholders in the selection of a "professional" management. They argue with some force that top management would be impelled to compromise group aims so as to anticipate and avoid conflict.

When, however, it becomes necessary to combine and direct the effort of two or ten thousand individuals, the primary unified purpose of the organization and its policies must be communicated to every individual who is charged with the exercise of discretion or with the co-ordination of the effort of subordinates.

The definition of aims and of policies is usually taken for granted and so receives no attention ; or when an attempt is made to define them, thinking is sometimes muddy and incomplete. To insist without qualification that the aim of business activity is the realization of profit is shallow. To aim at immediate profit is to produce policies and action far different from those which would be produced under an aim of long-time profit or under the so-called twenty-year point of view. The classic conflict between "production for profit" and "production for use" is understood only after we compare the consequences of decision and action in each case. The definition of purpose for non-profit units such as an educational institution, a hospital, or a chamber of commerce is an extremely difficult task, especially when one seeks the approval of many different minds in its acceptance. Then we discover that policies, like laws, qualify and limit purposes.

STATED AIMS AND ACTUAL PERFORMANCE

Even when the aim of activity has been clearly specified we must be sure that it is in fact the purpose which sparks all operations and control. Unfortunately, we can find numerous examples of obviously hypocritical presentation of aims in form for public consumption. Some group purposes, prepared for public consumption, glow with the warmth of interest in humanity ; but they turn out to be pure fiction when tested by actual performance and decision. It is not unusual to discover that the true purposes of a business unit, as evidenced by action, differ widely from professed purposes. If there is any suspicion that these remarks about purpose tend to overemphasize the obvious, it will be instructive and revealing to compare statements of purpose independently prepared by executives within an established organization.

In the pursuit of a specified aim, the performance of work is the aggregate performance of function by selected personnel in accordance with a method or system which dictates the time and place for such effort. In other words, the major concepts inherent in building, directing, and analyzing an organization of human beings are those of (a) function ; (b) personnel and (c) system. Numerous authorities offer many more so-called principles or concepts in organization which suggest more elaborate detailed treatment. But experience has already indicated that the more detailed classifications of ideas can and should be assembled as subdivisions under one of these three major classifications. Whenever this division cannot be followed, it will be found that the "principle" or "factor" refers to aim, to policy, or to some other standard.

TEST OF A "GOOD" ORGANIZATION

The test of a "good" organization then is the extent to which it achieves the purposes for which it was built, at a minimum cost. It is idle and irrational to judge an organization in terms of purposes which we would impute to it. We make many errors in estimating performance by our own standards when it should be measured by comparison with those actually used by the individual responsible for doing a job or supervising it.

So, in the direction of human effort, it becomes necessary to establish a sharp concept of "control." Let us define control as the continuous comparison of actual conditions or performance with standard conditions or performance, that is, a comparison of "what is" with "what ought to be." If all conditions and all performance conform with management's standards, management has no problems. This is to say that a problem exists, and is recognized, only when actual condition or performance fails to conform with accepted standard. In this sense then, the specific definition of a problem must include reference to the facts as they are, in comparison with the standards by which management is judging the facts of condition or performance.

It should be noted, however, that the phraseology in the preceding paragraph is intended to refer to comcomparisons with standards already established, or defined and accepted as a basis for comparison. Therefore it becomes necessary to point out that many management standards are not what they ought to be ; and therein lies another series of problems. Furthermore,

the standards which should be used are subject to frequent change. At this point then, management should become sensitive to the necessity of testing the adequacy of all standards at all times, and management should never assume that any standard should become rigid and not subject to change.

QUALITATIVE STANDARDS OF MANAGEMENT

It may be wise to remind ourselves that aims are one form of standard, policies are another, and the usual quantitative operating standards suggest another form. But we should also not neglect the definition of significant qualitative standards when they become pertinent to problem definition and to remedial action. Unconsciously, we are always seeking to measure phenomena ; and when we cannot measure them we make the mistake of ignoring or neglecting the qualitative influences in human affairs which are not measurable.

It should be acknowledged that the techniques in definition of qualitative standards call for special discussion because they usually entail special training. How can we, for example, define standards of loyalty, honesty, conscientiousness, intensity of effort, capacity to listen and follow instructions, or ability to sense significant demands of public-relations groups ? Qualitative standards are generally vague. They demand capacity in abstract thinking and in weighing emotional influences without the aid of measurement devices.

Now let us apply this control tool to the concept of organization. Failure in the performance of organization occurs when the functions being performed are not what they ought to be ; when the individuals in the organization are not producing as they should produce, and/or when the system of performance, especially with respect to timing, is not what it should be. Obviously, it becomes the responsibility of management to seek the reasons for variation of actual from standard in each of these three situations and to apply corrective measures.

ORGANIZATION ANALYSIS

Many organization charts are faulty because they have been conceived in confusion. The orderly presentation of the analysis of an organization requires a series of charts. The first chart should be one showing functional relations as they ought to be, without respect to personnel. The next chart should show functional relations as they are, without respect to personnel. A third organization chart should show personnel relationships as they ought to be, without consideration for the personalities or pressures of the individuals who occupy the defined positions. The fourth chart should show lines of authority and responsibility as they actually exist, with specific personalities occupying each position. If a comprehensive chart can be prepared which clearly includes all of these elements in the one presentation, so much the better. But the fear is that an adequate composite would simply promote confusion.

Other examples of the application of control suggest themselves readily. The usual operating ratios reflected in profit-and-loss statements and balance sheets are the obvious measures with which management is well acquainted and in the comparison of which most managements have already developed great facility. Yet it should be observed that even in this area judgement is flabby and indecisive whenever the appropriate standard has not been precisely defined. There are some situations entailing comparison with three or four different standards. According to one standard results are satisfactory, while according to another they are altogether unsatisfactory. Under these circumstances, it is essential that management select the standards which it seeks to meet.

REVENUE, COST, AND PRODUCTION ANALYSES

Sales quotas for territories, for products, and for individuals as well as groups are the most obvious standards in revenue analysis. Cost standards are widely used and well publicized but their interpretation and use have sometimes led to irregular action because their limitations were not understood.

Production control embodies the determination and use of thousands of standards of quantity, cost, and quality, in turn dependent upon the timing of the flow of internal and external traffic.

The estimate of an individual's or a department's contribution to performance can only be vague and useless unless there has been a reasonable standard of performance defined and understood by the individual as well as by his supervisor.

Conflict over estimate of quality arises frequently because inspectors have not been instructed properly to adopt a uniform standard. Numerous problems and costs have resulted from the failure to reconcile quality standards when they are dictated by different agencies which may have some jurisdiction over output.

Is there any administrator who has never had the experience of resorting to compromise to produce a single concept of what ought to be ?

EXTERNAL INFLUENCES

Discussion has been directed primarily toward internal controls. Although at any given time management may believe that its functions and its action are predominantly internal, it should be acknowledged that every decision or every activity may have associated external repercussions. Trivial internal details sometimes become magnified to produce substantial problems created by external pressure.

Management, as late as the turn of the century, took the position that it had the right to enforce standards of its own choice in the operations of its own enterprise. It may even be said that the institution of private property by definition included such rights. Also, it may even be that as we proceed to resolve some of the social conflicts of the past half century, we will be forced to the conclusion that vested rights rest upon the ultimate authority in defining standards.

In any case, current history reveals the extent to which group pressures, arising out of grievances against selected management practices, have promoted the crystallization of public opinion to such an extent that we now find the assumed right to define some of our own policy and operating standards seriously limited. The limitation becomes evident in the form of restrictive legislation or perhaps in the coalition of aggrieved groups for the imposition of economic sanctions or threats against the offender.

During the past four decades, and more particularly during the past two, groups external to management and affected by management's action have become vocal. Group organization has grown and is in the process of growing at a rate not readily appreciated by management. Through regulatory legislation, groups have imposed upon management standards which management did not or would not adopt voluntarily.

The force of group pressure becomes more apparent when we assemble the mass of human beings external to the business unit according to their respective functional interest in the unit being administered At the time a pertinent problem occurs in the usual business enterprise, management faces the demands and standards of customers, of creditors, of competitors, labour, government, stockholders and the general public. Each of these groups may be subdivided if more detailed classification promises to reveal more significant information which points more sharply to the standards and the conflict of standards within the major functional

groups. Creditors, for example, may be broken down into commercial banks, investment banks and brokers, merchandise creditors, and others with special claims against the assets. The general public may be subdivided by sex, by racial characteristics, by religious affiliation, by ages, or, for example, by expressed positions on particular issues.

If management could detect, define, and communicate the standards of these groups to its own organization and feel that the job had been done, there might be some comfort around the corner.

The fact is, however, that standards are always changing. The demands of labour, for example, are generally different today from what they were three or six months ago. So with the demands of customers, of creditors, of government. Here is the broad area of public relations extending far beyond the range of simple " publicity."

The most elusive skills in management are those prerequisite to the selection of the appropriate standard for comparison at a given time and to the detection of the need for changes in standard as well as to the reconciliation of those that are in apparent conflict.

MANAGEMENT MEASURED BY ITS STANDARDS

The skill of management, then, rests upon the use of adequate standards. They must be defensible before the bar of public opinion or they provoke regulatory legislation.

By this reasoning, government control appears predominantly in those functional areas in which the masses have suffered or expected to suffer maximum grievance. Wherever and whenever the masses felt justified in restricting action by management they have insisted through democratic processes upon the use of law to compel management's acceptance of standards dictated by the masses.

By this reasoning, government control has not appeared in those functional areas where management has been meeting mass standards.

In the latter case we say that management is regulating itself. A more precise meaning of self-regulation then is voluntary adherence to mass or group standards.

It would be careless at this point not to anticipate the serious question : Does this mean that private enterprise is destined to keep adjusting itself always to the temporary and inconsistent whims of the public ? The answer is a sharp negative.

We need to observe more than we do the influences which create mass standards. If we find them intolerable, and if we have a good case, we need to engage in positive compaigns to " educate " the masses to weigh the alternative offered by us. Here is a neglected activity. While management, representing capital alone, has been confining its energies to the protection of the rights of capital, friends of the masses have won their confidence and organized them to assert their own standards.

The issues over the designation of those best qualified to judge performance become academic; for finally, the choices and votes of affected external groups determine the standards of management.

Progressive management, then, is sensitive to the standards demanded by groups at interest. It recognizes that failure to effect compromises acceptable to groups at interest may bring increased government control; and it recognises that insistence upon the use of one's own standards rests upon the ability of management to convert groups at interest to the acceptance of those standards.

We must conform to the requirement of public opinion, or we must educate the public to prefer our standards.

SUMMARY

A concise summary, with great respect for language, may be helpful:

1. Management cloaked with specific authority, derived from a source originating in law, is responsible for the direction of human effort in the pursuit of well-defined aims.
2. An organization of human beings performs the functions necessary to achieve those aims; but it is limited by policy and system prescribed to produce coherence as a unit.
3. Control, based upon adequate definition of standards by management, reveals problems.
4. Decision by management estimates causes and provides remedial action.
5. But aims and policies as standards are determined not by management, but by the public, and are changing.
6. Regulatory legislation is the imposition upon management of standards which management did not accept voluntarily.
7. Management's course then becomes clearly a choice: (a) Accept public standards, or (b) oppose them and submit to legislative control, or (c) convert the public to the acceptance of management's standards.

Professional practitioners like engineers and physicians are surprised to discover that they are already skilled in these techniques. They need mainly to add the art of discrimination in selecting appropriate standards from a varying assortment. Scientific methods have not yet provided enduring bases for judgement in estimating human emotions. We still need alertness in clinical experimentation ; we need to manage our own minds, preserving an inexhaustible sense of humour.

AUSTRIA

Flow Distribution in Forced-Circulation Boilers

By M. LEDINEGG. (From *Maschinenbau und Wärmewirtschaft*, Vol. 3, No. 4, April, 1948, p. 49-57, 11 illustrations.)

WATER flow and evaporation are not necessarily uniform in neighbouring tubes connected in parallel, and special precautions are taken to ensure uniform parallel flow through the tubes of a water-tube boiler, e.g., individual throttling orifices are fitted at the inlet end of each steam tube of a La Mont boiler. Without precautions, the amount of water flowing through a tube depends on the rate of heat absorption. A small change in absorption rate may cause large changes in water flow and, under certain conditions, instability results, i.e., two values of water flow may exist for one value of heat absorption. In forced-circulation boilers, scale is sometimes formed at the individual tube orifices, whilst the tubes remain free of scale, and a restriction at the orifice follows, leading to reduction of flow and overheating. Assuming that a particle of water leaves the boiler drum at saturation pressure and temperature and arrives at the nozzle, after having passed the circulating pumps, still at the same temperature, it will evaporate after the nozzle if the absolute value of the pressure at the down-stream side of the nozzle is lower than the saturation pressure of the particle, i.e., the

pressure in the drum at the moment the particle left the drum. This may happen (a) during steady load, with extreme restrictions of cross-section in the orifice, or (b) if the drum pressure drops rapidly in the time interval which the particle takes to complete the circulation from drum to nozzle. Evaporation then leads to scale formation. A certain critical rate of pressure drop must not, therefore, be exceeded. In La Mont boilers, the rate of pressure drop is already restricted by the necessity of avoiding evaporation in the suction pipe of the circulating pump. However, the critical rate is smaller for the pipe nozzles than for the suction pipe, and this smaller critical rate is often exceeded in practice, and scale is then formed at the nozzles.

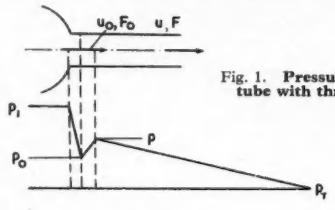


Fig. 1. **Pressure distribution in tube with throttling orifice.**

A. *The Throttling Orifice*

Fig. 1 shows pressure conditions at the nozzle. The pressure drops from p_1 before the orifice to p_0 in the latter, then rises to p behind the nozzle, due to the enlargement of cross-section, and finally drops to p_T in the drum owing to friction losses in the tube. Let F and F_0 be the cross-sections of the tube and the orifice, u and u_0 the velocities of flow, w_l the specific weight of water, C_c the contraction coefficient, G the weight of water flowing in unit time, then

$$p_1 - p_0 = \left(\frac{F}{C_c F_0}\right)^2 \frac{w_l u^2}{2g} = \frac{KG^2}{d_0^4 w_l} = C_1 \frac{G^2}{w_l} \quad (1)$$

K and C_1 are constants. The pressure rise, according to the law of conservation of momentum, is

$$p - p_0 = \frac{w_l u^2}{2g}\left(\frac{F}{C_c F_0} - 1\right) \quad .. \quad (2)$$

the overall pressure drop due to the nozzle

$$\Delta p_B = p_1 - p = \frac{w_l u^2}{2g}\left[\left(\frac{F}{C_c F_0} - 1\right)^2 + 1\right] \quad (3)$$

and

$$m = \frac{p_1 - p_0}{p_1 - p} = \frac{2\left(\frac{F}{C_c F_0} - 1\right)}{\left(\frac{F}{C_c F_0} - 1\right)^2 + 1} \quad .. \quad (4)$$

With $d/d_0 = 3$, from practice, and $C_c = 0{\cdot}6$, we obtain $m = 0{\cdot}15$, i.e., 15 per cent of the pressure drop is regained behind the nozzle. This pressure rise may reach, or exceed, the subsequent drop $p - p_T$ in the tube, so that at light loads, or if short and long tubes are connected in parallel in the boiler, $p_0 < p_T$, i.e., water may permanently evaporate at the nozzle.

B. *Rate of Pressure Drop and Conditions at the Nozzle*

Fig. 2 shows in full lines the pressure conditions during stationary working, and in dotted lines, the conditions after a drop $\Delta p'$ in the boiler pressure. Let us assume that during a time interval t, a particle of water circulates from the boiler drum through the pump and distribution header to the nozzle farthest from the header inlet, and that its temperature does not change during circulation and its saturation pressure, therefore, remains p_T. Assuming also that during

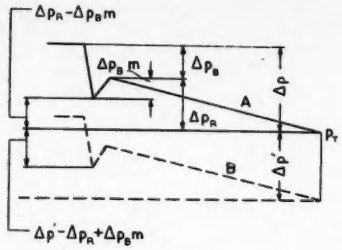


Fig. 2. **Pressure distribution with decreasing boiler pressure.**

time t, the boiler pressure has dropped by $\Delta p'$, then the absolute pressure at the nozzle has dropped below p_T by an amount shown in Fig. 2 and water will evaporate at the nozzle. If Δi_s is the change in total heat (B.Th.U. per lb) per unit pressure drop and L the latent heat, and if, further, the difference in static height, H, between the drum and nozzle at the tube inlet is neglected, the weight of (saturated) steam generated at the nozzle is

$$D_B = \frac{G\Delta i_s}{L}\left[\Delta p' - \Delta p_R + m\Delta p_B\right]$$
$$= \frac{G\Delta i_s}{L}\left[\Delta p' + m\Delta p - \Delta p_R(1 + m)\right] \quad .. \quad (5)$$

If steam formation in the orifice is taken into account in the calculations, the specific weight w of the steam-water mixture must be substituted for w_l :

$$w = \frac{1}{v_l + \frac{D_B}{G}(v_s - v_l)},$$

where v_s and v_l are the specific volumes of (saturated) steam and liquid. Neglecting the pressure rise behind the orifice, we obtain from eqs. (3) and (1) :

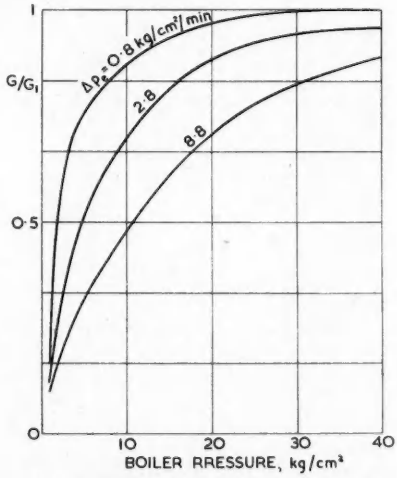


Fig. 3. **Change in weight o water circulating through a tube, as a function of the boiler pressure, for various rates of pressure drop.**

$\Delta p_e = \Delta p' t/60$ = rate of drop in boiler pressure, per min.
G = weight of water when evaporation takes place (see eq. (9)).
G_1 = weight of water, without evaporation in tube (see eq. (8)). Static head H is neglected.

$$\Delta p_B = C_1 G^2 \left[v_l + \frac{D_B}{G} (v_s - v_l) \right] \quad (6)$$

Assuming uniform heat absorption over the whole tube length l and a weight of (saturated) steam, D_0, at the tube outlet, we then obtain a mean specific weight w_m of the fluid in the tube,

$$w_m = \frac{1}{v_l + \frac{D_0}{2G}(v_s - v_l)}$$

and a pressure drop in the tube,

$$\Delta p_R = \frac{4 f l G^2}{2 g d w_m F^2} = C_2 \frac{G^2}{w_m} \quad .. \quad (7)$$

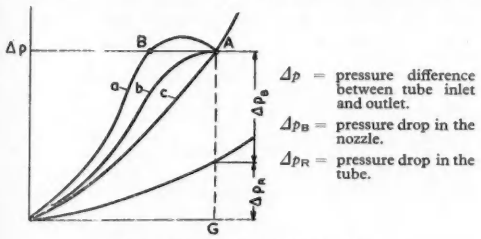
where f is a friction coefficient for losses in the steam tube and includes additions for bend and acceleration losses. The pressure drop from inlet header to drum, assuming there is no steam formation in the nozzle, is, from eqs. (1) and (7) :

$$\Delta p = \Delta p_B + \Delta p_R + H w_m = C_1 \frac{G_1^2}{w_l} + C_2 \frac{G_1^2}{w_m} + H w_m \quad .. \quad (8)$$

If steam formation takes place and the static head $H w_m$ is neglected, this equation must be derived from eqs. (6) and (7) with substitution for D_B from eq. (5), and transforms into

$$\Delta p = C_1 \frac{G^2}{w_l} + \frac{C_1 \Delta i_s G^2}{L} (\Delta p' + m \Delta p)(v_s - v_l) - \frac{\Delta i_s C_1 C_2 G^4}{L w_m} (1 + m)(v_s - v_l) + \frac{C_2 G^2}{w_m} \quad (9)$$

Assume Δp to be equal in both cases and assume, from practice, certain constant values for $\Delta p'$, then Fig. 3 demonstrates the effect of steam formation in the nozzle and shows that steam formation seriously reduces the flow only when boiler pressures are very low. However, deposition of scale at the orifice leads to a further, more serious, reduction due to the restriction of the orifice.



Δp = pressure difference between tube inlet and outlet.

Δp_B = pressure drop in the nozzle.

Δp_R = pressure drop in the tube.

Fig. 4. Instability at the nozzle.

That instability may occur at the nozzle is shown by the following considerations. Let $\Delta p'$ be constant and G decrease slightly, then Δp_R will decrease with G^2, eq. (7), and D_B will increase, eq. (5). With D_B, Δp_B must increase and, if steam formation is heavy, the increase of Δp_B may outweigh the decrease of Δp_R. The sum Δp may thus increase in spite of G falling. If the fall in G is more pronounced, it will outweigh the influence of increased evaporation and Δp will now decrease. This instability phenomenon is shown in Fig. 4, where A is the working point, curve a represents unstable conditions from A to B, and curve c stable conditions when, e.g., at higher working pressures, evaporation does not appreciably affect the flow in the tube. Curve b shows the limiting case at point A, the mathematical condition for which, from eq. (9) with $m = 0$, i.e., neglecting the pressure rise behind the nozzle, is as follows:

$$\frac{d\Delta p}{dG} = 0 = 2G \left[C_1 v_l + \frac{C_1 \Delta i_s \Delta p'}{L} (v_s - v_l) + \frac{C_2}{w_m} \right] - \frac{4 \Delta i_s C_1 C_2 G^3}{L w_m} (v_s - v_l) \quad .. \quad (10)$$

Instability is most pronounced when $D_B = 0$ at point A. D_B then increases most heavily with G slightly falling. From eq. (5), with $m = 0$, this is seen to occur for $\Delta p' = \Delta p_R$. If a boiler could be so designed that G, Δp_B, and $\Delta p_R = \Delta p'$ were constant for varying boiler pressure, then instability would occur, according to eq. (10), at working pressures below 150 psi only.

C. *Instability and Temperature in Distribution Header*

Under certain conditions of working pressure and feed water temperature, a different kind of instability occurring in forced-circulation boilers may lead to unequal flow in steam tubes. Fig. 5 shows the pressure drop in a heated pipe as a function of the fluid weight passing through the tube. Curve a represents conditions without evaporation taking place in the tube, and curves b, c, d conditions with evaporation, in order of increasing inlet temperatures. Curve b shows an unstable region D–A–B–C, and Curve d, representing higher inlet temperatures, is stable, Curve c gives the limiting case. Conditions according to Curve b must be avoided, even if the working range is below Δp_R, as load variations may reach into the unstable region. A lower limit is thus given for the water temperature in the inlet header, and the limiting condition is

$$\Delta i \leqslant \frac{v_l L C_3}{v_s} \left(1 + \frac{d}{4 f l} \right) \quad .. \quad (11)$$

where C_3 = const. and Δi is the difference between total heat of the water at tube inlet temperature and at saturation temperature. Δi mainly depends on working pressure and may be the greater, the higher the working pressure. If evaporation due to Δp_R, i.e., the pressure drop within the tube, is considered, conditions become slightly more unfavourable and for Δi must be substituted

$$\Delta i_U = \frac{\Delta i_s \Delta p_R + \Delta i}{1 + \frac{\Delta i_s \Delta p_R G}{W}} \quad .. \quad .. \quad (12)$$

where W is the rate of heat absorption. For normal working pressures, Δi_s is negligible and, therefore, $\Delta i_U \simeq \Delta i$. If throttling orifices are provided at the tube inlets, the stability condition is

$$\Delta i = \frac{v_l L C_3}{v_s} \left[1 + \frac{d \xi}{4 f l} \left(\frac{d}{d_0} \right)^4 \right] \quad (13)$$

where ξ is a factor accounting for jet contraction at the orifice.

D. *Instability and Rate of Heat Absorption*

In forced-circulation boilers, the drop in circulating water temperature is usually too small to cause instability, even if no nozzles are fitted. Nevertheless, nozzles are required because the tubes are heated to varying degrees according to their arrangement in the combustion chamber, and the heat absorption in different tubes may vary by more than 100 per cent. Since evaporation increases and the weight flowing through a tube decreases, with increasing rate of heat absorption, some tubes may receive an unduly small amount of water if heated too strongly. Introducing a stability criterion

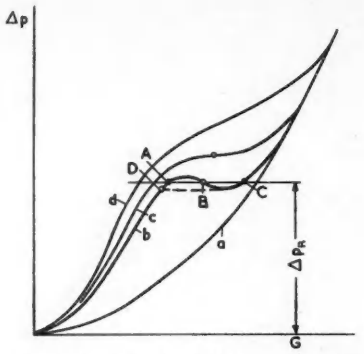


Fig. 5. **Pressure drop in a heated tube as a function of the fluid weight circulating through the tube, for different water inlet temperatures.**

$$Z = \frac{G}{W} \times \frac{dW}{dG} \quad .. \quad .. \quad (14)$$

indicating the relative change of fluid weight due to a certain relative change in rate of heat absorption, it is seen that the greater the change in G for a given change in W, i.e., the smaller Z, the greater is the tendency to instability.

The pressure difference between inlet header and drum of a forced-circulation boiler is

$$\Delta p = \Delta p_B + \Delta p_R + H w_m,$$

$$\Delta p = \frac{\xi G^2}{F_0^2 2g w_l} + \frac{4f}{2g d F^2}\left[l v_l G^2 + \frac{W l v_s}{2L}\left(G - \frac{2\Delta i G^2}{W^2} + \frac{\Delta i^2 G^2}{W^2} \right) \right] + \frac{H}{v_l + \frac{D_0}{2g}(v_s - v_l)} \quad .. \quad (15)$$

The first term of eq. (15) is taken from eq. (1) with slight modifications, e.g., the pressure rise behind the nozzle is neglected. The second term is derived from eq. (7) by considering that one part of Δp_R is used to add a total heat Δi to the water and bring it up to saturation temperature, whilst the remainder is used for evaporation only. The third term of eq. (15) accounts for the difference in static height between tube inlet and outlet. Fig. 6 represents eq. (15) for a certain given Δp and $H = 0$. Curve a, valid for $\Delta i = 0$, i.e., for circulating water entering the tube at saturation temperature, is stable. Curve c, drawn for lower water

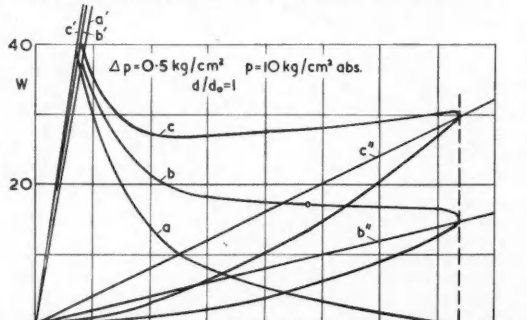


Fig. 6. **Rate of heat absorption as a function of the weight of fluid circulating through a tube, for different inlet temperatures below saturation point.**

temperatures at tube inlet (Δi finite), shows instability. Curve b is the limiting case. Lines a'', b'', c'' represent conditions when heat absorption is just insufficient to cause evaporation ($W = G\,\Delta i$), Lines a', b', c' represent conditions when heat absorption is so high that only dry saturated steam leaves the tube outlet ($W = G(\Delta i + L)$). Curves a, b, c have real significance only between their respective $'$ and $''$ lines. The maximum flow of water, G_{max}, is obtained when just no steam is generated and can be calculated from eq. (15) for $H = 0$ and $W = G\,\Delta i$.

Differentiating:

$$\frac{dW}{dG} = -\frac{\partial \psi}{\partial G} \Big/ \frac{\partial \psi}{\partial W}$$

where ψ represents the right hand side of eq. (15) (Δp = const.). Introducing eq. (14) and modifying, we obtain

$$\frac{v_s \Delta i}{v_l L}\left\{ \left[-Z - 1 - \left(\frac{\Delta i\, G}{W} \right)^3 (3 - Z) - K \right] \frac{W}{4G\Delta i} + 1 \right\} - 1 = \frac{\xi d}{4 f l}\left(\frac{d}{d_0} \right)^4 \quad .. \quad (16)$$

K accounts for the difference in static height:

$$K = H \frac{2gd\, w_m{}^2 F^2 (1 - Z)}{4 f l G^2} \times \frac{v_s - v_l}{v_s} \quad .. \quad (17)$$

G, W and Δi are known from designs and working data for the boiler. Z must be negative for stability, since, under stable conditions, W decreases with increasing G (see Fig. 6). Eq. (16) thus permits calculation of the minimum nozzle ratio d/d_0, i.e., the largest orifice compatible with a chosen safe stability value Z. The greater K, i.e., the static head H, or the greater the working pressure, or the smaller Δi, i.e., the nearer the inlet temperature to saturation point, the larger can the orifice be chosen for a given Z, and the more favourable, therefore, are the flow conditions. Fig. 7 shows the relative importance of these three component factors for $Z = -2{\cdot}7$. In existing plant, $\Delta p_R \simeq \Delta p_B$ and $-Z = 2{\cdot}5$ or larger.

E. *Oscillations of Steam-Water Columns*

If long tubes connected in parallel and open to the atmosphere at one end are supplied with a uniform water flow and are heated over their entire length so that steam is generated, the outflow is not always uniform, but sometimes shows periodic alternations of water with steam flow. Under certain conditions,* steam flow for 7 seconds was followed by water flow for 5 seconds, etc. Nozzles fitted at the tube inlets checked the oscillations and re-introduced stability. Steam locks which lead to rapid overheating of the boiler tube walls may be one cause for these oscillations. Not enough data are known for a rigorous theoretical treatment; however, by making many simplifying assumptions, the main causes of instability can be investigated. Assume that a steam lock has just broken up and that water begins to flow into the tube with an initial velocity v_1. The head of the water column cools the overheated tube walls and, apart from absorbing the normal transmitted heat, absorbs also the heat stored in the tube walls. Steam is generated, the water stream expands and the head assumes a much higher speed than v_1. The pressure at tube inlet rises because of friction and inertia effects, and may attain and exceed the pressure in the distribution header and thus stop further flow of water into the tube. The water already in the tube, minus steam that has been and continues to be generated, continues to stream, as a water column, to the tube outlet. Behind the water column, a new

* Vorkauf: Flow in steam generating tubes connected in parallel. *Archiv f. Wärmewirtschaft*, 1937, p. 75.

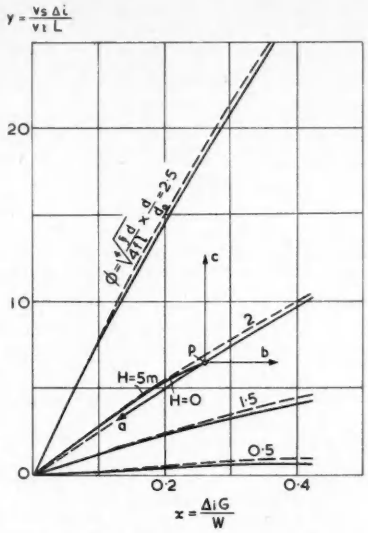


Fig. 7. **Relative influence of the parameters of equation (16), for a given constant margin against instability, $Z = -2{\cdot}7$.**

Dotted lines apply to $K = 0{\cdot}082$, corresponding, at 40 kg/cm² and under normal conditions, to a difference in static height between tube inlet and outlet of $H = 5$m; full lines apply to $K = 0$.

Assuming working conditions to be represented by point P, then an increase in water inlet temperature moves point P in direction a, a decrease in working pressure moves P in direction c, and a decrease in rate of heat absorption moves P in direction b. For a given Z, the coefficient ϕ for the nozzle ratio changes as shown.

steam lock forms and reheats the tube walls to higher temperatures. When the water column has passed the tube outlet, the pressure in the tube collapses and a new cycle starts. A condition for the development of these oscillations is a sufficiently high rise of pressure at the tube inlet. The tube length must, therefore, be above a certain minimum, and the steam lock must heat the walls to sufficiently high temperatures. For these conditions, the minimum tube length l_{min} to give rise to oscillations is given by

$$\left(\frac{d}{d_0}\right)^4 \frac{u_1 F \xi}{a(v_s - v_l)} = l_{min}\left(1 + \frac{4 f l_{min}}{2d}\right) \times \log_e\left(1 + \frac{2a(v_s - v_l) l_{min}}{u_1 F}\right) \quad (18)$$

a is a constant depending on actual tube length $l > l_{min}$. An increase of the nozzle ratio d/d_0 greatly increases l_{min} and, therefore, reduces the danger of oscillations. This equally applies to an increase in boiler pressure. Both tend to reduce instabilities of any kind.

F. *Conclusions for Boiler Design*

Evaporation at the nozzles can be avoided if feed water, suitably cooled, is introduced into the suction pipe of the circulation pump, provided the entire circulating water is cooled uniformly. If this is not possible, the pressure difference between the tube inlet and outlet should be increased by placing the distribution header as low as possible. The time required by circulating water to flow from the drum through the pump and header to the farthest nozzle should be kept as small as possible. Tubes connected in parallel should be of about equal length, as otherwise the shorter tubes must be fitted with smaller nozzles, which possibly introduce permanent steam formation. If the water temperatures at the tube inlets are much below saturation point, the margin of stability is reduced—even if instability from this cause cannot occur in La Mont boilers—and a slight increase of the nozzle ratio (see Fig. 7) would be advantageous. The water speed at the tube inlet should be high enough to carry away all steam bubbles; 3 fps is a safe practical speed if the margin against steam formation at the nozzle or instability is sufficient. If the nozzle ratio for the tube with the greatest pressure drop is so chosen that the pressure losses at the nozzle and in the tube are approximately equal, then the margin against instability is satisfactory. Other (shorter) tubes connected in parallel should then be fitted with correspondingly smaller nozzles.

HUNGARY

Grid-Controlled Switches for the Testing of Fuses with Alternating Currents

By P. V. Selmeci. (From *Elektrotechnika*, Vol. 40, No. 8, August, 1948, pp. 182-184, 3 illustrations.)

Fuse characteristics relate the fuse blowing and clearing time to the current carried by the fuse. These characteristics are the basis for the design of selective protection systems for networks.

The characteristics give a clear picture of the fuse when the applied current is constant. This condition is only partly fulfilled by direct current, as the current reaches the constant value, not instantaneously, but after a certain time, which depends on the relation between the inductance and the resistance of the fuse circuit. Characteristics obtained in this way are valid only for this type of fuse in other circuits if their inductance-resistance ratio is the same. Often there is no direct source of sufficient power and voltage available to carry out tests on fuses, so that, therefore, the idea of using alternating currents suggests itself.

The current applied from an alternating current source has its transient values when applied to the fuse circuit. These transient current values depend on the circuit constants and the closing time of the circuit. In the case of very large currents, the fuse will blow before the transients die off or even before the current reaches its maximum value. This last property is an important advantage of fuses.

In actual circuits the transient current connects the steady state of the circuit with that following the short-circuit. If the short-circuiting is instantaneous, the current increases to a value determined by the short-circuit impedance of the circuit, unless the fuse blows before reaching the maximum current. When the short-circuiting is not instantaneous, consecutive half-cycles of the current show increasing maxima and the fusing occurs at a later period. The greater the ratio of circuit resistance to inductance, the more quickly the current increases. Generally, when a fuse is loaded by its maximum current rating, the fuse melts during the first increasing quarter-cycle of the current and clears during the second quarter-cycle. These two wave components are non-sinusoidal and unequal.

These considerations point to the importance of the correct choice of closing time of the fuse circuit when testing the fuse with a.c. The inductance and the

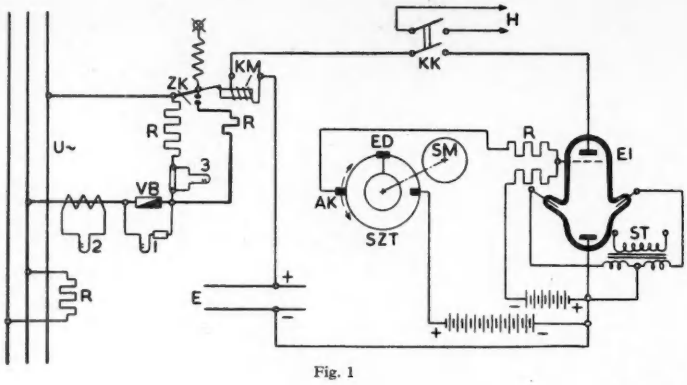


Fig. 1

resistance of the circuit are known. To be able to obtain the maximum current determined by these circuit constants, the circuits will have to be switched on when the current is just passing zero. A special switch is required for this purpose. Its accuracy should be $\pm$ 0·5 millisecond, or $\pm$ 10 electrical degrees.

This type of switch is called a synchronous switch. The simplest solution would be to use a synchronous motor with a cam shaft, the cam switching on at the appropriate moment.

The synchronous switch which has been used for the tests is shown in Fig. 1. Its action is indirect, i.e., by means of a thyratron. The grid voltage is negative when the switch is off. A synchronous motor *SM* rotates the contactor *ED*. When the contactor reaches the brush *AK*, the second grid circuit is closed. The grid becomes positive and the thyratron ignition closes the circuit of the solenoid *KM*. The core of the solenoid is attracted and the switch *ZK* closes the circuit of the fuse *VB*. The time of closing is changed by the relative displacement of the brushes *AK*.

The synchronous motor has four poles and, therefore, the position of the contactor *ED* relative to the current wave, has to be known to avoid position errors by 90 electrical degrees. A white line is painted on a rotating drum on the motor shaft and illuminated by a glow discharge lamp. The lamp is fed by a 110-volt d.c. source on which a 110-volt a.c. source is superimposed, thus flashing the lamp 50 times per minute. When the position of *ED* is correct, the white line appears.

Fig. 2 shows an advantageous alternative of a synchronous switch. The synchronous motor is replaced by a pentode. The pentode grid is connected to an induction regulator *ISZ*. The output of the pentode feeds the thyratron grid. By altering the position of the motor of the induction regulator, the switching-on time of the fuse circuit is changed.

1, 2 and 3 are oscillograph points. Fig. 3 shows oscillograms of fuse tests, where 1, 2 and 3 are the graphs of the relative oscillograph points. 1 shows the voltage on the fuse, 2 the fuse short-circuit current and 3 the steady current before the short-circuiting on a highly magnified scale. The tests were carried out on 100 amp. nominal "Binet" fuses. The resistance *R* was chosen to allow a maximum current

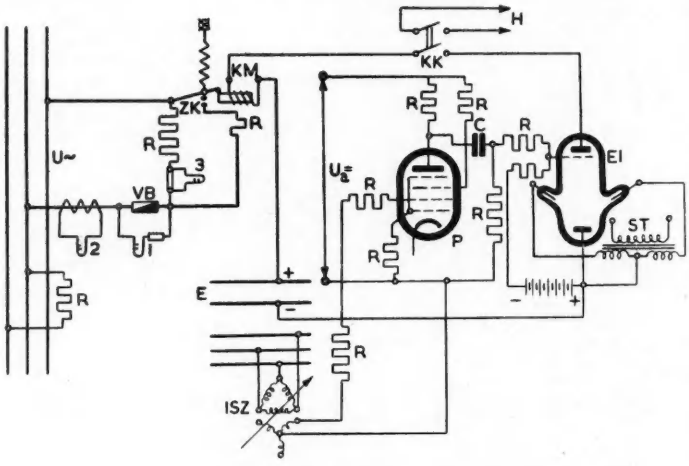


Fig. 2

intensity of 7000 amp. when the voltage was 400 V.

As shown, the fuse circuit was closed at different points of the voltage cycle and the fuses withstood the load. The working of the synchronous switch is satisfactory. Further experiments are being carried out and will be reported when results are available.

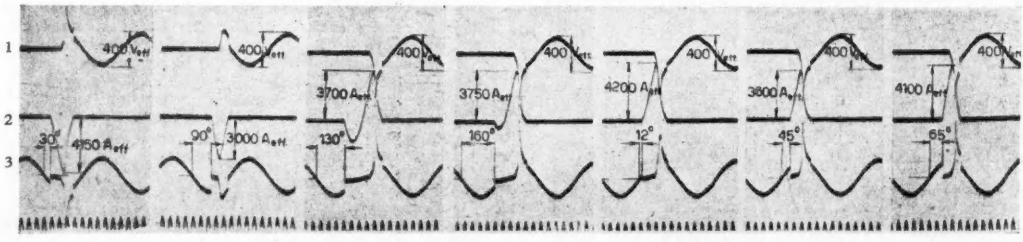


Fig. 3

Smooth Regulation of Spot Welding Sets by means of a Choke in their Secondary Circuit

By A. Z. BLITSTEIN. (From *Avtogennoye Delo*, No. 10, 1947, pp. 6-8, 5 illustrations).

A METHOD is proposed for the control of welding processes which serves the same purpose as the usual transformers with tappings, but is considerably simpler. Very close regulation has been proved possible by introducing into the secondary of the transformer a choke with a variable air gap.

$$I_1/I_2 \approx L_2/M \quad .. \quad .. \quad (1)$$

where L_2 is the inductance of transformer secondary and M the mutual inductance.
Leakage inductance

$$L_{s_2} = L_2 W_2/W_1 \quad .. \quad .. \quad (2)$$

This, introduced into the first formula, results in:—

$$I_1/I_2 \approx W_2/W_1 + L_{s_2}/M \quad .. \quad .. \quad (3)$$

which clearly indicates the possibility of regulating the welding current by means of leakage reactance. It has been proposed by the author to use a magnetic core around the lead carrying the welding current; the air gap of the core is varied in order to obtain regulation of the welding current (Fig. 1). The secondary of the welding circuit acts as the winding of the choke.

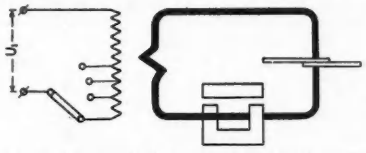


Fig. 1. **Magnetic core in the secondary circuit of a welding machine.**

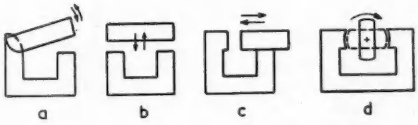


Fig. 2. **Methods of varying the air gap of the magnetic core.**

If the welding current is to be regulated from I_2 to I_2*

$$I_2 = \frac{U_{2o}}{Z_2} = \frac{U_{2o}}{\sqrt{R_2^2 + X_2^2}} \quad .. \quad .. \quad (4)$$

where R_2 = resistance of the secondary circuit;
X_2 = inductance of the secondary circuit;
U_{2o} = No-load voltage in secondary circuit.

If Z_2 is increased to Z_2* by means of the secondary choke

$$Z_2^* = \frac{U_{2o}}{I_2^*} = \sqrt{R_2^2 + (X_2 + x_{ch})^2} \quad .. \quad (5)$$

and from this

$$x_{ch} = \sqrt{Z_2^{*2} - R_2^2} - X_2 \quad .. \quad .. \quad (6)$$

The drop in the secondary voltage caused by the choke

$$\Delta U_{ch} = I_2^* x_{ch} \approx 4{\cdot}44 f W_2 \phi_m$$

The maximum magnetic flux is, consequently,

$$\phi_m = 4{\cdot}5 \times 10^{-3} \Delta U_{ch} \quad .. \quad (7)$$

The cross-section of the magnetic core is determined from the maximum flux.

The air gap can be varied by any one of the methods shown in Fig. 2. The proposed method of regulation

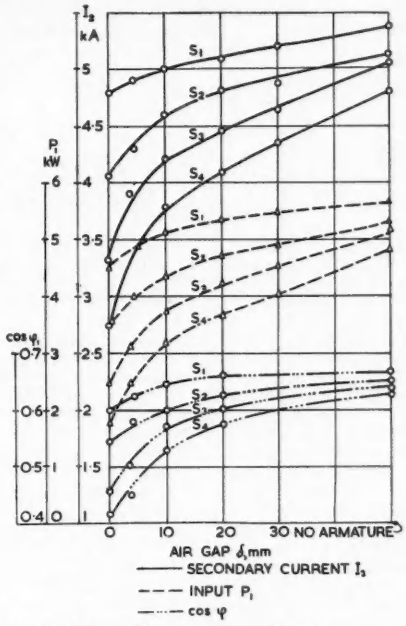


Fig. 3. **Variation of I_2, power input P_1 and cos φ_1 as a function of the core section and air gap.**

was experimentally carried out on a standard type spot-welding machine. Fig. 3 represents the experimental results, namely, I_2, power input P_1 and cos φ_1 over the airgap δ for four different cross-sections of the magnetic core, i.e., $S_1 = 10$ cm², $S_2 = 15$ cm², $S_3 = 20$ cm², and $S_4 = 25$ cm².

Without the magnetic core, the results were:—

$I_2 = 6450$ A;
$P_1 = 7600$ W;
cos $\varphi = 0{\cdot}745$.

The range of regulation can be gauged from Fig. 3.

The experiments have proved that the examined spot welder lends itself to smooth regulation from

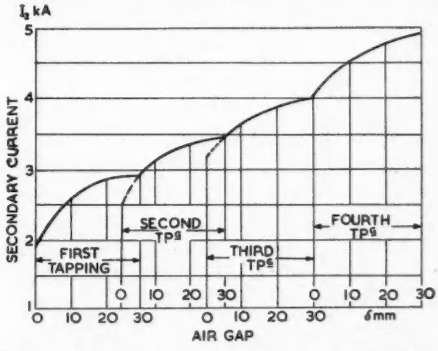


Fig. 4. **Variation of secondary current I_2 with different transformer tappings and changing air gap in the magnetic core.**

5000 A down to 2000 A with a cross-section of the iron core of 15 cm² and using a standard transformer with four tappings.

A further advantage of the magnetic core is a steeper characteristic of the machine and the fact that the no-load secondary voltage remains constant if the leakage inductance is increased. This permits welding with a high secondary voltage and low current—a useful condition for certain applications such as welding of insufficiently cleaned parts. The magnetic core with an armature constructed as shown in Fig. 2d can also be used as a current interrupting device for roller spot-welding machines as it is found to be sufficient to reduce the current by 60-70 per cent instead of complete interruption.

The main advantage of the proposed method of regulation consists in its simplicity and cheapness and, at the same time, in its wide range. The smooth regulation obtained is particularly important when spot or seam-welding large pieces of ferromagnetic materials, the moving of which in the secondary circuit may otherwise lead to an uneven quality of weld.

ITALY

New Transatlantic Ships

By L. CALZAVARA. (From *Rivista Marittima, Technical Supplement*, July, 1948, pp. 138-159, 2 illustrations.)

A LARGE number of passenger ships were destroyed during the war, so that thought will have to be given in the near future to the rebuilding of the world's passenger fleets.

Shipbuilders are thus faced with the problem of drawing up a program which will take into account present circumstances and future possibilities. For this purpose, various factors regarding economy and design require consideration, which include, in particular, the following:—

(1) As sea transport will never be able to compete with air transport as regards speed, the speed of the new ships should not be increased beyond a certain limit, for instance, 23-25 knots.
(2) Furthermore, some of the first-class passengers will be travelling by air; therefore, the luxury accommodation may be reduced while increasing accommodation for tourist-class passengers.
(3) The loss in revenue from passengers can be compensated by developing the freight transport services, bearing in mind that freight rates may decrease considerably later on.
(4) The technical means of making ships more economical in operation should be carefully studied, particularly in connection with (a) increasing the number of passengers and providing more comfortable accommodation; (b) increasing the freight capacity; (c) reducing the fuel consumption by improved hull shapes and more efficient propulsive machinery.

It was found under normal pre-war conditions that the large high-speed luxury liners frequently sailed with only 30-40 per cent of their passenger capacity, and therefore, in spite of the high fares charged, did not succeed in covering their expenses. On the other hand, ships of lesser tonnage, such as the "Oceania" and the "Neptunia," which had a lower speed and were more economical to operate, gave excellent financial results. Consequently, it may be anticipated that future ships will have a displacement of 20,000-30,000 tons, with speeds between 20 and 25 knots.

Swimming pools should be abolished, since they are not only non-hygienic, but also carry the risk of upsetting the stability of the vessel.

To increase the space available for passengers, two alternative methods can be adopted, viz., either by designing for lateral loading of cargo, making it possible to develop the superstructure more towards the forward deck, or by reducing the space taken up by the propulsive equipment, particularly as regards height.

The space taken up by the power plant in large ocean liners is nowadays approximately 190-200 m³ per 1,000 hp for steamships, approximately 250-280 m³ per 1,000 hp for motorships with a single engine-room, and approximately 500-550 m³ per 1,000 hp for motorships with several engine-rooms.

A reduction of 50 per cent in the machinery space would enable over 200 extra passengers of the tourist class to be carried in a mixed motorship of 25,000 tons and 25 knots. This increase in revenue would pay for all the fuel requirements of each journey. Furthermore, a saving in engine weight per shaft horse-power should also be achieved, so as to obtain, for instance, a reduction from 75 or 80 kg/hp to 30-40 kg/hp for steamships, and from 60 or 100 kg/hp to 30-40 kg/hp in motorships. This would correspond to a saving of about 2,700 tons on a 25,000-ton vessel. It would appear that steam propulsion, in spite of recent and past progress, will not be able to fulfil the above-mentioned requirements, in view of the space required for boilers and exhaust-gas ducts, and the higher specific consumption. Attention should, therefore, be particularly concentrated on Diesel engines; the gas turbine and the atomic engine have not yet appeared on the horizon, although the gas turbine may offer considerable possibilities. To reduce weight and save space, higher-speed Diesels can be used in conjunction with an electrical, hydraulic, or gear-type power transmission. The use of transmission gears results in a considerable saving of weight and cost as compared with electrical transmission, and has been tried out by German engineers. The engine should preferably be a horizontal H-type with two pairs of opposed cylinders, giving a particularly compact design, and balancing arms can be used to connect the four connecting-rods to either one or two crankshafts situated below or on either side of the cylinders.

Another subject of particular importance nowadays is the transport of emigrants from Europe to North and South America and elsewhere, as a result of which many ships leave Europe with full passenger loads, but do not find passengers for the return journey. The ships are, therefore, used as cargo ships on the return journey. The present arrangement of setting up and removing passenger berths takes up too much time and berths are frequently damaged in the process. Suggested solutions would be to use collapsible berths which can disappear into recesses in the walls or deck floors when not in use.

Further suggestions refer to navigational safety. For reasons of economy, shipbuilders generally design their ships with the propulsive machinery in a single room or compartment. In the event of damage or a breakdown in the engine room, however, this means that the ship will be completely immobilized. The possibility should, therefore, be examined of arranging auxiliary electric generators and motors in a separate compartment. These auxiliary units would supply the electric energy required for lighting, etc., under normal conditions, and could be used for propulsion under emergency conditions, giving the ship a reduced speed of 6-7 knots. The use of non-inflammable materials should, moreover, be extended as far as possible in new ships.

DESIGN AND WORKSHOP

A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.

Gearing

HOW TO REDUCE NOISE AND INCREASE EFFICIENCY WITH MODIFIED TOOTH FORMS

By K. N. Mills. (From *Machine Design, U.S.A.*, Vol. 21, No. 1, January, 1949, pp. 88-92, 8 illustrations.)

High efficiency and low operating-noise level of modern gearing is the result of increased generating accuracy, and increased knowledge and application of the principles responsible for quiet gear operation. These principles cover the design of gear teeth, shafts and gear housings and, if the optimum results are to be obtained, each of these factors must be given minute attention.

In the case of bevel, spur and single helical gears some of the adverse effects of housing and shaft deflection have been eliminated by introducing a slight curve into the tooth profile in the longitudinal plane. As this type of tooth profile modification tends to concentrate the tooth load in the centre of the tooth, it decreases the possibility of the tooth load being concentrated on the ends of the teeth in the event of excessive housing or shaft deflection. However, as it limits the tooth bearing area, it increases the tooth contact stresses beyond the stress level which would exist if ideal contact conditions were maintained.

Because the surface durability of gear teeth is a function of the magnitude of the subsurface contact stresses, " crowning " the tooth face causes a reduction in the theoretical load capacity of the gear train. If the shafts, housing and gears were absolutely rigid this theoretical loss in capacity would be real. These elements are not always rigid and deflections always exist. For these reasons crowning of gear teeth may result in lower actual contact stresses because the degree of stress concentration due to deflection may exceed that induced by the crowned-tooth profile.

All modern high-efficiency gear trains use the involute gear tooth because it has certain advantages not found in other systems. The gear-tooth profile is described by the end of a generating line rolling on the base circle, and it can be visualized as a curve traced by unwinding a string from the circumference of the base circle. As the involute is generated from the base circle, it cannot extend below this circle. If any portion of the tooth profile extends below the base circle, that portion of the tooth profile will be inactive and must be generated on an assumed profile. This portion of the tooth profile will be undercut as shown in Fig. 1, causing a reduction in the tooth beam strength.

Action of a mating pair of involute gears can be compared to that which would occur if two pulleys with diameters corresponding to the base circle diameters were connected with an endless crossed belt, the straight sections of which represent the lines of action for both directions of rotation. The point at which they cross represents the pitch point or the point at which the pitch circles are tangent. This example illustrates one of the distinctive characteristics of the involute system of gearing, which is its flexibility in centre spacing.

It is obvious that, if the centre distance is altered, the obliquity of the straight portions of the crossed belts with the line of centres would change. Also the location of the intersection of the crossed belts and the line of centres would shift. However, the speed ratio of the assembly would remain constant. Similarly, the centre distance of a pair of engaging involutes can be varied within certain limits without altering their speed ratio. When this change is made in a pair of generated involute gears, the pressure angle and running pitch diameters are modified, as the diameter of the base circle is unaltered.

Due to the flexibility of the generated involute system of gearing, it is possible to alter the outside diameters of engaging gears to eliminate undesirable features such as undercut and to obtain more advantageous tooth action. In speed-reduction drives, this is usually accomplished by increasing the outside diameter of the pinion and decreasing the outside diameter of the gear. Gears thus proportioned are cut to standard depths and operate on standard centres. Because the addendum on the pinion is larger than the standard addendum and the addendum on the gear is shorter than the standard addendum, this is called the long and short addendum system of gearing.

A comparison of the tooth action obtained with an undercut pinion and a gear train with long and short addendum teeth is shown in Fig. 2. In this illustration, the pinion tooth strength has been increased as the thickness of the tooth at its base has been increased. Also, the length of the active portion of the line of action has been increased. Therefore, the number of teeth in contact has been increased. This increase in the number of teeth in contact reduces the load carried by each tooth and reduces the noise level of the gear train.

In the case of the long and short addendum gear train, it will be noted that the lengths of the arc of approach and recession are not equal and the arc of approach has been materially shortened. This change in the length of the arc of approach decreases the possibility of the tips of the driven teeth digging into the flanks of the driver teeth under heavy loads and scraping the oil film away in front of the contacting teeth. During recession, the tips of the driver teeth have a wiping

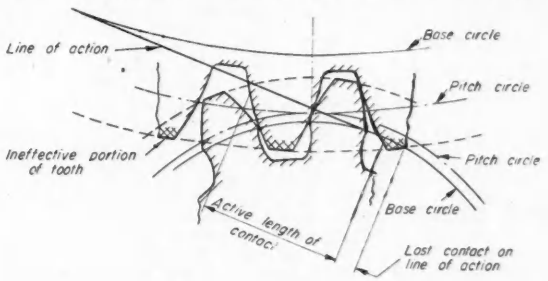


Fig. 1. **Undercut of tooth form to provide clearance reduces its strength.**

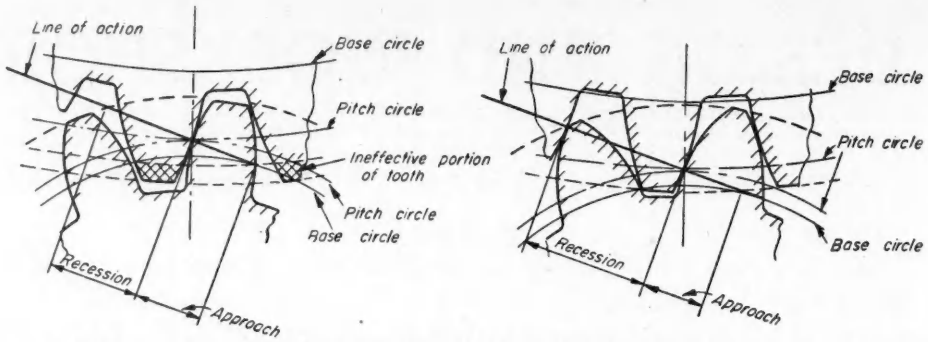


Fig. 2. **Standard gear teeth compared with long and short addendum modifications, increasing the strength of the pinion. Modified gear is shown at right.**

action which is conducive to smooth action and good lubrication.

That portion of the involute adjacent to the base circle is difficult to generate accurately; therefore, the accuracy of the pinion teeth is increased because they are further removed from the base circle. Also the specific sliding between the teeth is reduced because it is very high on that portion of the involute near the base circle and reaches infinity there.

Involute gears can be designed for operation on centre distances greater than their theoretical centres. This result can be achieved by any one of the following three methods:

(1) Enlarging the outside diameters of both the driver and driven elements;

(2) Enlarging the outside diameter of the driver and using a standard driven element;

(3) Enlarging the outside diameter of the driven and using a standard driver element.

Gears used in spread-centre applications can be produced with standard cutters, if they are cut by the generating process and the teeth are cut to standard depth or enlarged blanks. When the enlarged gears are generated, their base circle diameters are not altered, because they are fixed by the base circle diameter of the generator cutter. However, the pressure angle is altered because the centre spacing between the cutter and the cut element is changed and the pressure angle is a function of the angle between the common tangent to the two base circles and the line of centres.

A layout of a pair of gears designed to operate on spread centres is shown in Fig. 3. In this illustration, it will be noted that the theoretical pitch circles are not tangent, and the gears mesh at running pitch circles which are larger than their theoretical pitch circles.

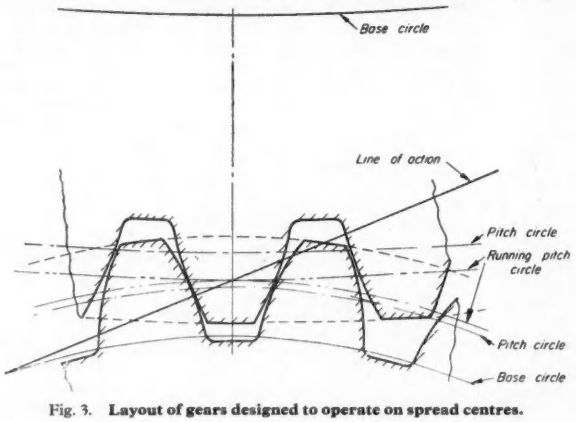


Fig. 3. **Layout of gears designed to operate on spread centres.**

When designing gears for operating on spread centres, the width of the tooth tip should eb checked on both elements. The minimum tooth tip width should not be less than 10 per cent of the tip width of a standard tooth.

Enlarging both the driver and driven elements to produce gears for use on increased centres should be employed, if the drive is reversing. It may be used on non-reversing drives when the pinion is not undercut and the pinion tooth strength is satisfactory. In this system, the relative enlargements should be proportioned so that the lengths of approach and recession are equal. This result can be achieved by making the distance from the running pitch circle to the outside diameter of the gear blank equal on both elements.

The system of enlarging the driver and using a standard driven is preferable for speed-reducing drives as it retains all of the advantages of long and short addendum gearing. However, it should not be used for speed-increasing drives. This practice is advantageous when three or more gears are in mesh and the intermediate gear functions as both driver and driven. In this case, the third gear can be cut with standard outside diameter and tooth proportions. This will maintain the maximum length of line of action.

Using an enlarged driven and a standard driver should only be employed to eliminate undercut. When used to eliminate undercut it does not adversely affect the length of the line of action, the undercut portion of the tooth being inactive. However, because undercut decreases tooth strength, a stronger tooth and improved tooth action will be obtained.

SOURCES OF NOISE

Noise level of a gear train is a function of the accuracy of tooth spacing and profile as well as tooth deflection and number of teeth in contact. The adverse effects of spacing and profile errors can be partially nullified by introducing a slight amount of relief to the tip of the tooth as shown in Fig. 4. The effect of this tip relief is to permit the driven gear to lag slightly and thus pick up any errors gradually. This decreases the possibility of the driven teeth digging into the flanks of the driver teeth, and scraping the oil film from these teeth. If relief is applied to the driven teeth only it will prevent interference between the tips of these teeth and the sensitive portion of the involute near the base circle.

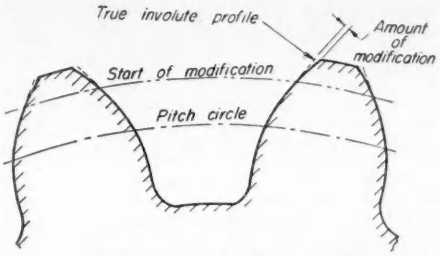


Fig. 4. **Relief on tooth tip reduces effect of errors in tooth spacing and profile.**

While this practice has a number of theoretical advantages in spur gearing, the value of these advantages in single and double helical gearing is questionable. Because that portion of the tooth beyond the start of modification is cut below the involute profile, theoretically it is inactive. Therefore, the theoretical length of the line of action is decreased as shown in Fig. 5. The amount of this reduction is dependent on the extent of the tip relief and the contact lengthening effect of tooth deflection. A small amount of tip relief will produce a material reduction in the theoretical length of the line of action.

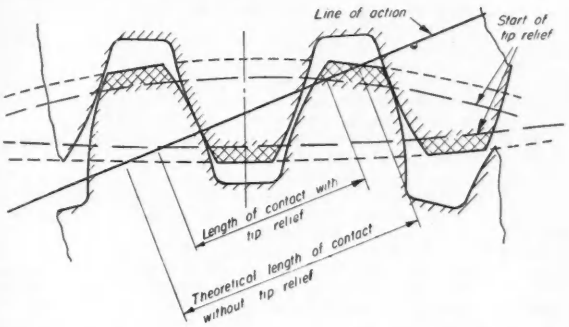


Fig. 5. **Effect of tip relief on reducing the length of the line of action.**

In well proportioned single and double-helical gear trains, tooth deflection is extremely small due to the multiple tooth contact obtained and the common use of the stub tooth form. Also, in this type gearing, the edge of the tooth contacts first, and the load is gradually applied to the tooth. If long and short addendum gears are used, the effect of deflection is further reduced because of increased strength of the pinion teeth, and the reduction in engaging impact resulting from the shortened arc of approach.

Amount of modification produced in a tooth by a given cutter is dependent on the number of teeth to be cut in the gear and the type of the gear teeth. The modification effected by a given cutter will decrease as the number of teeth in the gear is decreased. A given cutter will produce more tip modification on a decreased diameter gear than a standard diameter gear. For this reason, gears with tip modification may be more expensive to produce because separate cutters may be required for the gear and pinion. If tip modification is used in a gear train, a thorough investigation should be made of the tooth action, because excessive modification can cause tip bearing on pinion teeth.

One of the major factors influencing the quietness of a gear train is the number of teeth in contact and the variation in total length of tooth contact in single and double-helical gear trains. In a spur gear train the number of teeth in contact should not be less than 1·4.

EFFECT OF ELASTIC DISTORTION

While the basic characteristics of helical gearing promote smooth tooth action, elastic distortion of the teeth can introduce slight variations in tooth action. If it were practical to design all gear trains so that the number of teeth in the plane of rotation was slightly greater than an integral number, this effect would be materially reduced. However, this is not always practical, but helical gear trains can be designed so that the variation in total length of tooth contact is a minimum.

If a gear train is designed to ensure the maintenance of this condition, the adverse effect of these small tooth deflections will be kept at a minimum. Whilst the result cannot be achieved in all gear trains without using special cutters having non-standard pressure angles, it can be achieved or approached by adjusting diameters, diametral pitches and face widths in some cases.

If the average number of teeth in contact in the axial plane is slightly greater than an integral number, there will be no variation in the length of tooth contact. If a gear train is designed to meet either of these conditions, the resultant gear train will have a very low operating noise level.

Electrolytic Tool Sharpening *

By S. E. Noskov. (From *Stanki i Instrument, Russia*, Vol. 19, No. 10, October, 1948, pp. 20-22, 4 illustrations.)

The method of electrolytic tool sharpening or grinding developed by Gussev, allows of sharpening tools of any degree of hardness, regardless of their mechanical properties, on a grinding wheel having a considerably lower degree of hardness than the tool to be sharpened. At the same time, the geometrical form of the tool is preserved and a good finish on the tool faces (absence of cracks and pitting) is assured. The whole process of sharpening, from the rough cut to the finishing cut, is performed on the same wheel and with the same setting.

The cost of electrolytic sharpening is considerably reduced by the possibility of substituting a copper or steel disk, easily made in any workshop, for the usual high-grade grinding wheel.

This method employs direct current and a special electrolyte. The tool to be ground and the grinding wheel are connected in a d.c. circuit so that the tool forms the anode while the cathode is formed by the grinding wheel (Fig. 1). By the action of the direct current in the electrolyte, metal is continuously removed from the surface of the anode, i.e., the tool being treated.

The inventor of the electrolytic process claims that the removal of metal proceeds both by electrochemical and mechanical action on the anode. According to the theory advanced by the inventor, the passage of a direct current through the special electrolyte used causes the formation of a film of insoluble compounds on the anode, protecting it from further electrolytic action. Removal of this protective film from the surface of the anode by the rotating grinding wheel forming the cathode exposes a fresh surface of the metal to the action of the electrolyte and, thus, assures the continuous removal of metal from the anode.

* Abstract by courtesy of Industrial Diamond Information Bureau, 32-34, Holborn Viaduct, London, E.C.1.

THE ESSENTIAL PARAMETERS OF THE ELECTROLYTIC GRINDING PROCESS

The efficiency and quality of electrolytic grinding depend essentially on correct selection of the electrical parameters. The higher the current intensity and voltage, i.e., the greater the quantity of electrical energy supplied to the grinding point, the greater the quantity of metal removed in unit time, and, consequently, the greater the efficiency of the process. On the other hand, the greater the current density, the greater the roughness, i.e., the lower the quality of the resulting surface. Three stages of the process are provided for: the roughing cut, coarse grinding, and finishing. Each stage has a corresponding voltage, current intensity, and degree of finish.

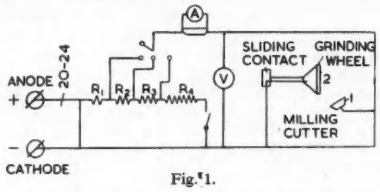


Fig. 1.

The numerical values corresponding to these stages are summarized in Table I :

TABLE I.

Stage	Tension volts	Current amps	Metal removed mm/min	Surface roughness
Roughing ..	18–20	40–45	0·5–2·0	2·5–6·4
Coarse grinding	17–19	20–30	2·3–0·8	0·5–2·5
Finishing ..	10–14	2–4	0·03–0·06	0·13–0·5

During grinding, sparking at the anode and fluctuation of the instrument needles are slight; in the finishing stage, all such external signs are practically absent.

The efficiency of the electrolytical grinding process is considerably influenced, not only by the electrical values, but also by the speed of the cathode, the pressure of the cathode on the anode, the rate of feed of the electrolyte, and the concentration of the latter.

Cathode Speed: A low speed of the cathode promotes shorting and causes roughness of the surface. With the cathode running at a very high speed, the electrolyte is forced out of the electrode gap by the centrifugal effect and electrolytic action ceases. In ordinary work, the grinding wheel should run at 8-12 m/sec.

Cathode-Anode Pressure: With increasing pressure of the cathode on the anode, the rate of removal of metal increases to a peak value, and then falls abruptly, despite further increased pressure. It is recommended that the unit pressure be maintained within the limits of 0·4-0·8 kg/cm². Fig. 2 shows the relationship between rate of removal of the metal and cathode pressure, voltage, and corresponding current intensity.

Electrolyte Feed: To ensure smooth progress without interruption, the electrolyte should be continuously fed to the grinding point in suitable quantity. Supply from a gear or centrifugal pump with a delivery of 6-10 litres per minute is sufficient.

MACHINE FOR ELECTROLYTIC GRINDING

Electrolytic grinding can be performed on ordinary grinders, suitably adapted. No industrial model of a special anodic grinder has yet been developed but for introduction of the process, an adaptation of the UTZA 64 standard grinder of the " Ilyitch " Works is recommended. The following assemblies on the " Ilyitch " Works grinder have been re-designed.

The headstock assembly has been modified, the standard headstock being replaced by a special design, adapted for anodic grinding (Fig. 3). The grinding wheel is carried on one end of the spindle (1), the other end of which is bored to receive a spring (2). The spindle is movably keyed on a sleeve forming a pulley (3), and carries a fixed contactor ring (4), through which the current is taken from the brushes to the spindle and grinding wheel. The sleeve in which the spindle moves is mounted in ball-bearings in the headstock. The grinding wheel is fed forward and pressed against the anode (the work) by means of the aforesaid spring, and rotated by the pulley already mentioned. The grinding wheel rotates at 1420 rpm and has a diameter of 150 mm.

A special 50-litre tank for lubricant, tundishes, piping and nozzle have been designed for this grinder.

The entire electrical control equipment of the grinder (resistors, instruments, starters, fuses and switches) is mounted on a separate column. The electrical control system is shown diagrammatically in Fig. 4.

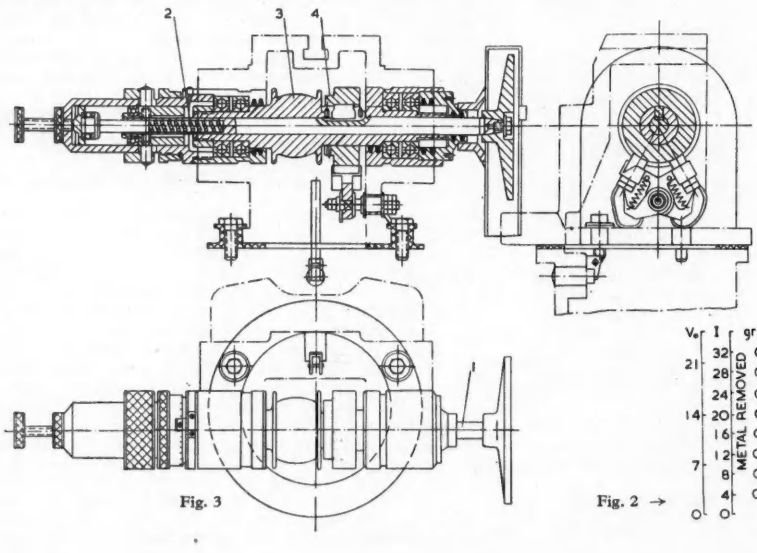
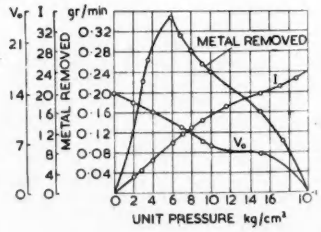


Fig. 3

Fig. 2 →



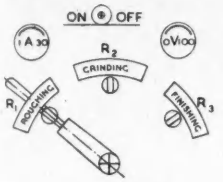


Fig. 4.

The current intensity is regulated by moving the controller from the roughing to the grinding and finishing terminals successively, thus changing the corresponding resistances. The resistance values are: for roughing, $R_1 = 0{\cdot}05 - 0{\cdot}1\ \Omega$; for grinding, $R_2 = 0{\cdot}25\ \Omega$; and for finishing, $R_3 = 1{\cdot}8\ \Omega$; $R_4 = 3{\cdot}6\ \Omega$.

For grinding lathe tools and cutters, the required nominal voltage of the d.c. generator is $E_{nom} = 24$-25 volts; $I_{max} = 50$ amps.

The d.c. supply for electrolytic grinders may be taken from generators with a stable external characteristic, i.e., capable of supplying a steady voltage under variable load. The most suitable type for the purpose is a low-voltage galvanising generator for 6-12 volts.

To obtain the voltage necessary for electrolytic grinding, the generator should be run at 1420 rpm.

In the absence of a galvanising generator any of the following may be used:

(1) *Multiple Welding Generators.* These can be most effectively used for simultaneously operating a number of grinders.

(2) *Aircraft Generator Sets.* These are particularly compact and convenient for mounting on the machine.

(3) *Tank Generators*, of 1000-1500 watts output.

(4) *Selenium, Copper Oxide or Mechanical Rectifiers.*

Comparative data on sources of direct current are given in Table II.

TABLE II.

D.C. Source	Rated Voltage	Current Amps	Power consumption kW	Efficiency	Weight kg	No. of coupled grinders
Galvanising generator	24	250	11·25	0·53	400	4–6
Multiple welding generator	25	750	18·5	0·7	350	6–7
Aircraft generator	24	36	2·0	0·5	22	1
Tank generator	25	70	3·0	0·6	30	1–2
Mechanical rectifier	25	50	2·2	0·5	50	1
Selenium and copper oxide rectifier	25	100	—	0·9	100	2

The World's Largest Testing Machine

(From a booklet commemorating the first official demonstration of the new 5,000,000-lb Testing Machine at the Philadelphia Naval Base on 29th September, 1948.)

IN the Aeronautical Structures Laboratory at the Philadelphia Navy Yard, there is a testing machine which makes it possible (1) to measure the loads which the largest and strongest aeroplane structural elements will withstand; (2) to modify their design and minimize weight with assurance that any unforeseen and unpredictable concentrations of stress will be detected and eliminated, and (3) to solve the principal problems of designing full-scale aircraft on the basis of small-scale model design and tests.

One of the objectives of tests of large full-size structures will be to correlate them with similar tests of scale models in order to determine so-called "size effects." In other words, the purpose of the correlation is to make model testing a more accurate basis for design.

Tests of structures to-day yield far more than a stress-strain curve ending at the break of a test specimen. By means of SR-4 bonded resistance wire strain gauges on aircraft structures, panels, beams, etc., in testing machines, the Navy's Aeronautical Structures Laboratory will determine stresses at as many points as may be required to show where added strength may be required or where material may be removed to reduce weight. These measurements will be made at any measured loads below and approaching the breaking point of the test specimens. Such stress measurements will be made rapidly with the aid of automatic equipment which will record 48 gauge readings in 12 seconds.

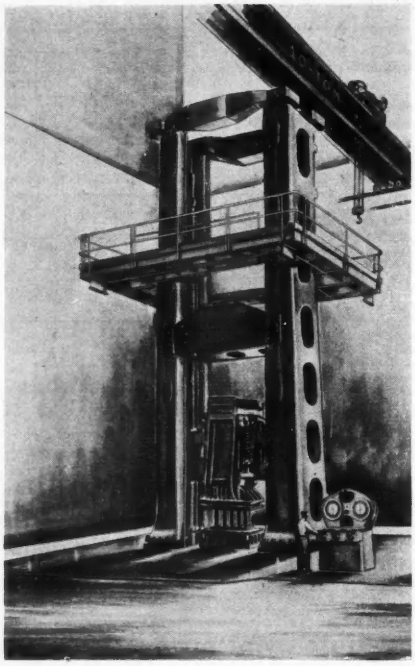
The new 5,000,000-lb testing machine is of the same design as other recently-built testing machines of two to four million pounds capacity. Specimens to be subjected to compression or flexure are mounted on the bed plate at floor level or on a levelling table on the bed plate. Tensile test specimens are held between the sensitive crosshead and the tensile test crosshead above it. The tensile test crosshead is rigidly held on the columns at one of several positions spaced above five feet apart.

Load is applied downward by the sensitive crosshead, which is supported on the two 55-ft screws, which are in turn supported by arms cast integral with the hydraulic loading cylinder under the bed plate. The weight of this assembly, close to 250,000 lb, is floated on two auxiliary hydraulic cylinders when the machine is not in use. These cylinders are bolted to the main cylinder casting. They serve also as shock absorbers when a test specimen breaks. The hydraulic loading cylinder can move 36 in. downward whereas the piston is stationary, being secured to the bed plate. Hydraulic loading pressure is applied by a 75-hp electrically-driven pump of 3000 psi delivery pressure which produces testing speeds up to 3 ipm. Return speed is 15 ipm. The sensitive crosshead is positioned by a 50-hp motor rotating the screws by means of drive gears under the bed plate. The sensitive crosshead also serves as an elevator for the tension crosshead.

The load-measuring system consists of the Emery cell, carried by the sensitive crosshead, and a Tate-Emery indicator with two indicating dials and accessory equipment in a cabinet beside the machine. This cabinet also contains operating controls. Emery cell and indicator are connected only by a flexible tubing for transmission of hydraulic pressure.

The Emery cell, which transmits load pressures to the indicator, is essentially a shallow cylinder having a loose fitting piston and a metal diaphragm so arranged that the load on the specimen produces a pressure change on a thin film of liquid trapped between the piston (and diaphragm) and the bottom of the cylinder. Relative movement of piston and cylinder is practically negligible.

The flexible tubing from the Emery cell ends in a Bourdon tube in the indicator. The Bourdon tube does not actuate the indicator pointer on the dial directly, but tends to move a baffle plate over an airjet in a pneumatic system, thus producing a substantial



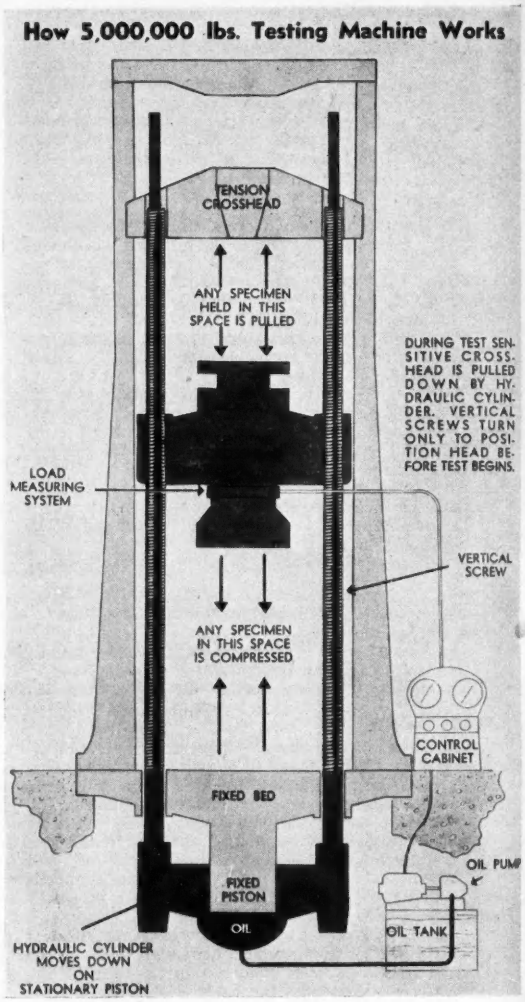
change of air pressure in this system. This resulting change of air pressure actuates a sensitive bellows which exerts an opposing force against the Bourdon tube movement, thus maintaining the Bourdon tube and baffle in a substantially fixed position. This opposing force is applied through highly accurate load-measuring springs, the extension of which is magnified by an indicator pointer on a 66-in. scale on the dial. Six ranges are provided by six Bourdon tubes and air-jet baffles, and two bellows servo-motors and dials.

The testing machine, made by the Baldwin Locomotive Works, is essentially a hydraulic press. However, the necessity for accurately measuring applied loads, often off centre, is an important consideration in design. Three design features maintain minimum tolerance in horizontal deflections under load, prevent premature buckling of compression specimens, and neutralize horizontal components of force. These are (1) crossed flex plate fulcrums or stays between the sensitive crosshead and its sensitive yoke; (2) adjustable guides for the sensitive crosshead, and (3) flaring of columns at the base.

The crossed flex plate construction permits the slight relative vertical movement necessary to compress the Emery cell between crosshead and yoke but prevents horizontal movement of the yoke in any direction. Flaring the columns stiffens them and enlarges the bases to 6 ft. by 8 ft., permitting the use of 21 two-inch diameter bolts to secure each to the massive bed plate.

One of the largest testing machine accessories is an elevator consisting of a platform surrounding the machine. This platform is suspended from the upper horizontal frame member on wire cables. A 15-hp motor under the bed plate raises and lowers the elevator at 12 fpm. The elevator may be controlled at the control cabinet or on the platform.

There are a number of safety features incorporated in the design of the machine to prevent misuse of and damage to it. For the safety of the operator the control and indicator cabinet is located at one side where tests


How 5,000,000 lbs. Testing Machine Works
TENSION CROSSHEAD
ANY SPECIMEN HELD IN THIS SPACE IS PULLED
DURING TEST SENSITIVE CROSSHEAD IS PULLED DOWN BY HYDRAULIC CYLINDER. VERTICAL SCREWS TURN ONLY TO POSITION HEAD BEFORE TEST BEGINS.
LOAD MEASURING SYSTEM
VERTICAL SCREW
ANY SPECIMEN IN THIS SPACE IS COMPRESSED
CONTROL CABINET
FIXED BED
OIL PUMP
FIXED PISTON
OIL
OIL TANK
HYDRAULIC CYLINDER MOVES DOWN ON STATIONARY PISTON

can be watched and controlled most conveniently. The machine has also been equipped with a safety screen to prevent heavy parts of breaking specimens from being thrown away from the machine.

Many of the compression tests to be made by the Aeronautical laboratory require extremely close dimensional tolerances on flatness and parallelism between loading surfaces. For this purpose a special bearing plate 72 in. square is attached to the upper sensitive weighing head. Parallelism of this plate is adjusted by rotation of the plate on slightly tapered surfaces.

In some tests it is desirable to maintain a constant rate of increasing load. For this purpose the machine is equipped with a load rate pacing disk and associated automatic controls. In other tests it is desirable to maintain a constant rate of straining the test specimen. A similar strain rate pacing disk and associated automatic controls will hold the strain rate constant. A motion indicator is also supplied for the sensitive crosshead. In addition, the machine is equipped with a self-contained, automatic, autographic recording apparatus for making load-elongation records on charts.

Hard Chromium-Plated Bearing Surfaces with Anti-Friction Qualities

By K. GEBAUER. (From *Metalloberfläche, Germany*, Vol. 2, No. 8, August, 1948, pp. 161-165, 11 illustrations.)

LUBRICATING oil does not adhere to hard chromium-plated bearing surfaces. Wear due to friction is, therefore, especially severe at low speeds (starting-up dead-centre position in reciprocating machinery) and high pressures. The poor adhesion of oil can probably be attributed to thin, invisible and very hard oxide layers which also affect the special surface qualities of hard chromium-plating. Anti-friction qualities may be slightly improved by using special lubricants, by increasing play, by decreasing bearing pressure, and by selecting appropriate bearing metals to suit the chromium-plated surface. The wetting qualities of the plating can only be improved at the expense of other qualities, e.g., coarse-crystalline chromium plating has excellent wetting qualities but is softer than a normal hard chromium deposit and shows insufficient cohesion of the single crystallites (so-called porous chromium.) As a result of this observation it was proposed to roughen the chromium surface, i.e., to break up the bearing surface with small " oil grooves " (or pockets) which help to re-establish the broken oil film. This idea was successfully applied during the war to cylinder liners for Diesel engines. The grooves can be cut or etched prior to chromium-plating (Fig. 1) or can be etched out after plating (Fig. 2). The advantages of the first method are that the chromium thickness is independent of the desired groove depth, that the chromium adheres equally well inside the grooves and to the bearing surface, that the plating is of uniform thickness and can withstand appreciable wear without adversely affecting the efficiency of the oil grooves. The latter method requires thicker plating since the grooves can only be worked out of the deposit, and is more expensive, since finishing takes longer.

The grooves and pockets are so small that they are

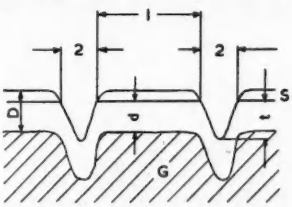


Fig. 1. **Schematic cross-section of bearing surface with indentations produced mechanically *before* plating.** Chromium layer can be thin, depth of " oil pockets " is independent of thickness of chromium layer, little or no finishing work required. (For explanations see Fig. 2).

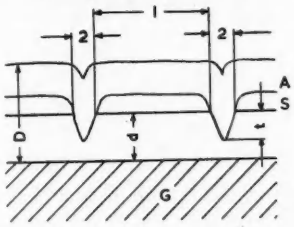


Fig. 2. **Schematic cross-section through bearing surface with oil pockets produced by etching *after* plating.** Thick chromium layer, depth of oil pockets dependent on thickness of layer, which is reduced by etching and grinding. (G = parent metal, D = thickness of deposited chromium layer, d = thickness of deposit after finish machining, A = reduction of layer thickness by etching, S = reduction by grinding, 1 = bearing surface after grinding, 2 = loss of bearing surface caused by " oil pockets," t = depth of " oil pockets.")

more like indentations ($0{\cdot}15$ to $1{\cdot}5 \times 10^{-3}$ sq in) or scratches ($0{\cdot}5$ to 6×10^{-3} in width). The total recessed area occupies from 2 to 20 per cent of the total bearing area and should be larger, the greater the risk of disruption of the oil film under the bearing pressure. A high-duty four-stroke petrol engine was run satisfactorily with a 10 per cent recessed area. A uniform distribution of "oil pockets" is essential for good functioning; in some cases, only highly stressed zones of the bearing need be roughened, e.g., the dead-centre zones in I.C. engine cylinders. The depth of the recesses determines the permissible amount of wear and varies between about 10^{-4} to 5×10^{-3} in. The maximum thickness of the deposit is only 6×10^{-3} in. The efficiency of the pockets can be increased if wetting agents such as colloidal graphite or engine carbon deposits are firmly embedded in the recesses.

If recesses are formed *before* plating, they can be etched in by pickling or by an electrolytic process. Since pickling attacks the basic material fairly uniformly, long pickling periods are required, the material loses its dimensional accuracy and must be ground and honed very carefully in order to re-establish a smooth parent surface without removing or flattening the etched recesses. This method is, therefore, expensive and, because of the flatness of the resulting recesses, is not very effective. A mechanical method, lower in cost, consists in sandblasting with subsequent grinding and honing. This method produces only flat recesses and can be applied whenever high dimensional accuracy is of no consequence. Another method consists in " knurling " the finished and honed surface. Distribution, area and depth of recesses can easily be controlled by the shape of the knurling tool which usually forms conical or pyramidal recesses. The parent surface is honed again to remove slight burrs at the edges of the recesses and can then be chromium-plated to the required thickness and dimensional tolerance. No further finish is required for dimensional accuracy, but a final honing process is sometimes applied to remove plating nodules. Many cylinder liners for petrol and Diesel engines have been successfully treated in this way. Carbon is deposited in the recesses and lubricating oil consumption is reduced to normal after only a few hours running-in. Yet another method consists in turning into the parent surface a fine helical groove of 0·004-in depth and about 0·1-in pitch and honing before plating in order to remove the ridges formed by turning.

Preparation *after* plating can only be done chemically, since chromium is too hard and brittle for normal mechanical recessing. A network of hairline cracks is distributed over the hard chromium deposit, its relative coarseness varying with electrolytic conditions. Baths with SO_4 as catalyst produce the coarsest network, SO_4/Si F_6 baths a medium, and F/Si F_6 baths the finest network. The hardness and brittleness of the deposit are greater, the finer the network. These cracks, hardly visible at a magnification of 120, can be widened and deepened by pickling in hydrochloric acid or in hot dilute sulphuric acid, with or without the use of electric current, and by anodic or, preferably, cathodic treatment. The smooth chromium surface is attacked less when cathodic treatment is adopted. Alkaline liquids can be used for etching in anodic treatment. The edges of the cracks are attacked first, and the cracks thus widened. After a longer treatment, the undamaged surface is also attacked, starting at the cracks, so that finally a surface with numerous irregular recesses is obtained after grinding and honing.

The recesses are fairly deep, and such surfaces are already usable for light speeds and pressures. The method is costly, since a thick chromium layer has to be deposited, this layer then being partly dissolved and finally ground to produce the necessary bearing surface. Since the parent metal and the deposit absorb hydrogen, which reduces their strength, it has been customary to apply heat, after plating, to a temperature of 200° C in order to diffuse the hydrogen. Because of the difference in thermal expansion between the parent metal and the chromium, a network of " heat " cracks is produced in the deposit, with coarser and wider cracks completely penetrating to the bottom of the deposited layer. These cracks are etched, and after subsequent grinding and honing, a bearing surface with fairly wide and deep oil grooves is obtained. Etching need not be as prolonged as for the " natural " cracks. This method is, therefore, cheaper and more effective because of the greater depth of the cracks and because oil is retained by capillary attraction.

The different methods can be combined as required. Normal hard chromium-plating baths and methods can be applied, although special solutions are used to give fine or wide mesh crack networks. The grinding disks and honing stones should have a fine grain and soft binder (*K* to *L*, Norton scale).

Thorough cleaning of the roughened surfaces is essential if the fine " oil pockets " and " grooves " are to be effective.

New Goods Locomotive 1-5-0

By L. M. SYTYCH. (From *Vestnik Machinostroyenia, Russia*, No. 2, 1948, pp. 29-30, 1 illustration.)

THE new locomotive has been developed and built at the Kolomensk Locomotive Works, and while it is 30 tons lighter than another new Russian locomotive, its hauling power is practically the same. It has been equipped with a stoker and central lubricating system for the axles. The large diameter of the superheater tubes ensures a fairly constant steam temperature of 380-410° C. The boiler is of all-welded construction, which results in a considerable saving in weight as compared with rivetted boilers. Satisfactory welds are ensured by the use of automatic welding control and X-ray examination. The fire box is of the radial type with a 6 m² fire-grate. The fuel distribution is even, due to stoker firing, and this produces more intensive combustion, higher temperatures and a reduction in fuel consumption.

A high-speed steam engine is fitted on the tender, which drives a screw conveyor for coal through a reduction gear. The steam pressure in the engine varies between 0·5 and 2 kg/cm², depending on the fuel. The steam and coal pipes connecting the engine and tender have spherical couplings and the coal supply pipes are of telescopic design. The conveyor brings the coal up to the opening of the fire-box, from which it is blown on to the fire-grate by steam.

A steam dryer is placed in the steam dome. The regulator is of a single-valve type with a pilot valve.

Two oil pumps ensure the reliability of the central lubricating system for the steam distribution as well as for the axles.

The cylindrical steel block of the engine consists of two parts. Cast iron liners are pressed into the steel cylinders, this being necessary as the piston rings as well as the distributing slide valve rings are of cast iron. The diameter of the cylinders is 650 mm and the piston stroke 300 mm.

All bearings of the steam distributing mechanism are of the needle type. This reduces wear as well as the weight of the parts and, thus, facilitates the balancing of the locomotive.

In order to reduce wear induction hardening has been widely applied.

The tender weight is distributed over two bogies of the wagon type.

Exhaustive tests on this locomotive have been carried out and have given very satisfactory results. Its fuel consumption is found to be 24 per cent lower than that of other types of the same power. Its haulage characteristics are superior and the required maintenance and repairs are less than that of comparable types of engines.

Technical Characteristics.

Total weight of locomotive	103 tonnes
Adhesion weight of locomotive	91 tonnes
Rating	2100-2200 hp
Hauling power	19200 kg
Design speed	80 km/hr
Overall length	13800 mm
Steam pressure	14 kg/cm² gauge
Area of fire-grate	6 m²
Boiler tube surface	222 m²
Superheater surface	131 m²
Weight of loaded tender	82 tonnes
Loading of tender axle	20·5 tonnes
Water capacity	28 tonnes
Coal capacity	18 tonnes

Drilling Bush Attached to Portable Drills

By M. REICHNER. (From *Werkstatt und Betrieb, Germany*, Vol. 81, No. 6, June, 1948, p. 163, 1 illustration.)

DRILLING small holes with a portable drill often results in drill breakage. Fig. 1 shows a simple jig which can be fitted to electrical and other portable drills and has been proved to prevent frequent drill breakage. It can be used generally, but is uneconomical where the drill has to be exchanged frequently. The enlarged end of tube *d* is attached rigidly to the frame *f* of the portable drill. The small end of this tube *d* is screwed on the outside and carries the cylindrical box *c* which is locked in position. The thread is long enough to

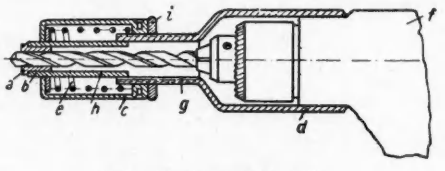


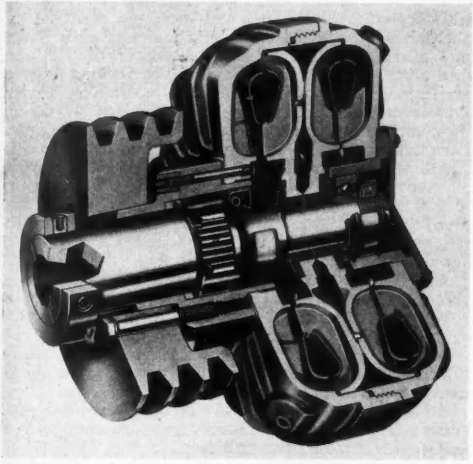
Fig. 1. Jig for portable drill.
a = drilling bush, *b* = sliding tube, *c* = cylindrical box, *d* = screwed and enlarged tube, *e* = spring, *f* = portable drill, *g* = keyway, *h* = key, *i* = locking nut.

ensure adjustment of box *c* to suit different lengths of drills, e.g., after frequent re-grinding. Box *c* contains spring *e* which presses against the flange of tube *b*. Tube *b* can slide inside the small end of tube *d* (but is prevented from turning by the keyway arrangement *g*, and is kept in its outermost position by the pressure of spring *e*. The hardened bush *a*, finally, is pressed into tube *b* and is used to guide the drill. As the drill penetrates more deeply into the work piece, tube *b* and bush *a* slide backwards against the pressure of the spring. The drill is thus guided next to the work piece during the entire drilling process. The jig described is especially useful where, due to restricted space, long drills have to be used. The neck of tube *d* can be made longer with only a very small loss in rigidity of the entire jig.

New Materials, Processes and Equipment

HYDRO-SHEAVE DRIVE

The Hydro-Sheave Drive designed by Twin Disc Clutch Co., Rockford, Ill., is a hydraulic-conversion device for small electric motors and internal combustion engines. The basis of the new unit is a Twin Disc Small Hydraulic Coupling, to smother shocks and smooth the flow of power from motors or engines in the ¾ to 25 hp range. Oil is the only medium of power transmission. Incoming power from the electric motor or internal combustion engine rotates the hydraulic coupling inner member, the blades of which

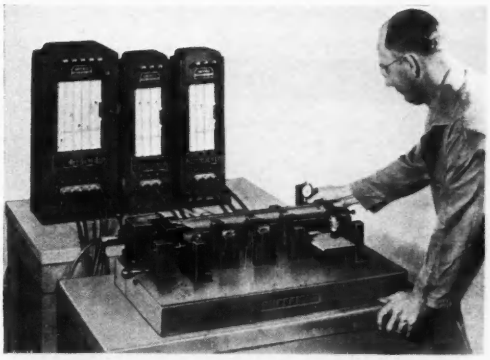


throw a continuous stream of oil against the vanes of the coupling outer member, or housing. The rotation of the housing with its attached sheave or sprocket provides a steady, uninterrupted flow of power to the driven equipment. A sudden shock, or overload, from the driven equipment is absorbed by the fluid in the hydraulic coupling and never reaches the motor or engine. Conversely, torsional vibrations from the engine or motor are effectively damped out by the fluid and never reach the driven equipment.

CAMSHAFT PRECISIONAIRE

A machine which permits a complete and almost simultaneous gauging of most critical dimensions and conditions of an automobile camshaft has been developed by The Sheffield Corporation of Dayton, Ohio. Six bearing diameters and two lengths of the camshaft are checked with airsnaps used in conjunction with a five-column and a three-column Precisionaire base instrument. The shaft is revolved manually to check for out-of-round, and also for concentricity of five bearings which is indicated by a variation in float position in an additional three-column Precisionaire. Checking of two lengths is accomplished by additional air jets.

Run-out of the oil pump drive gear is checked by a separate gauging unit consisting of a mating gear mounted on a ball slide and actuating a dial indicator. The relief hole is checked for clear passage by playing a jet of air into the end, the stream of air being felt by the operator when the passage is unrestricted. A Scleroscope is mounted in front of the front gauging



station to determine hardness while the camshaft is in gauging position.

ABRASION TESTING SET

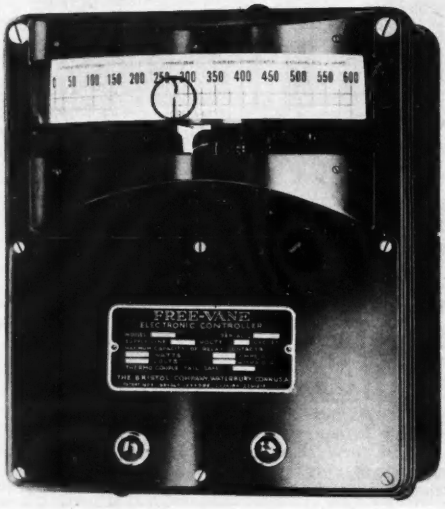
With this new Abrasion Testing Set manufactured by Taber Instrument Corporation, North Tonawanda, N.Y., the wear results from alternately rubbing the flat faces of two resilient Calibrase wheels over the surface being tested. A range of standardized abrasive wheels is available for testing all types of surface finishes, including electroplate, porcelain enamel, organic coatings, leather, glass, plastics and woven textile fabrics.

New features include improved suction pump with vertical dust receiver with swinging nozzle fully adjustable for any thickness of specimen. The control panel is equipped with an adjustable timer to provide a test period of 1000 cycles, or less, and then automatically shutting off the power permitting the operating technician to be otherwise engaged in tabulating results or preparing new specimens without concern of overrunning the test being made.

An important feature of the new testing set is that the control unit is equipped for dual operation, that is, an additional Abraser unit can be located beside and connected with the control unit, thereby permitting two specimens to be tested simultaneously using the same controls and vacuum unit.

PROPORTIONING PYROMETER CONTROLLER

The development of a proportional current-input electronic pyrometer controller has been announced by The Bristol Company, Waterbury 91, Conn. The new instrument proportions the current input to electrically-heated furnaces, ovens, plastic moulding machines, salt pots, and other similar equipment to provide

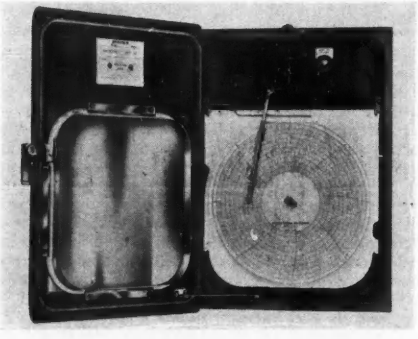


practically straight-line temperature control. It does this by time modulation of the input energy. The average energy supplied is proportional to the deviation of the temperature from the control point throughout a band width, which is adjustable from 0 to 2½ per cent of full scale reading.

RUNNING COUNT RECORDING INSTRUMENT

A new recording instrument known as the Running-Count Recorder has just been announced by The Bristol Company, Waterbury, Conn. The newly developed recorder plots on a circular chart a curve of number of operations against time. It records the total count of intermittent operations and the time at which each one occurs, and is used on production machinery to record work-producing operations or number of pieces produced.

With this Bristol Recorder, the total number of operations or pieces produced can be easily found by



multiplying the number of complete pen tranverses across the chart by the count per tranverse for which the instrument is calibrated.

The hourly rate of production can also be easily read from the chart. This gives information regarding effect of fatigue on operators, variations in productibility between operators, effectiveness of job training programs, and the effect of variations in working conditions. Excessive down time is easily read on the chart. The instrument is also used as a basis for paying production employees. It furnishes a true and accurate record upon which to base payment.

IN THIS ISSUE—*continued from page* 71.

In the first two or three decades of this century, expert opinion was mainly based on the publications of F. W. Taylor as they were then understood and possibly misinterpreted. His pupils and successors were mostly technicians whose education and training were based on "pure science," such as mathematics, physics and chemistry, and who were accustomed to drawing reliable conclusions on this firm foundation. Although Taylor himself repeatedly mentioned the necessity of taking physiological and psychological factors into consideration when dealing with human labour, the knowledge in this sphere was still too limited or too inaccurate to avoid the many mistakes which characterised the management of that time. A reaction was inevitable, which had already started before the last war, but which gathered momentum throughout its duration, and this is reflected almost all over the world.

Social science, the doctrine of mutual relationship among human beings and of relationship between human beings and material elements of their environment, was pressed into the foreground, and management had to become acquainted with ideas following totally different lines from those of the pure and technical sciences previously mentioned.

In actual fact, a change of mentality was necessary, which was not simple as this new mentality entered into conflict with training and tradition.

However, as can so often be observed in similar cases, and as is reflected to a large extent in present-day literature, the pendulum is swinging too far in one direction, the technical aspect of management problems nowadays being neglected and many publications concerned with management reading like philosophical or even religious pamphlets.

Without underestimating this tendency in any way, it is important, in these circumstances, as in many other cases, to advocate the happy medium. The reader who may not be accustomed to this kind of literature will perhaps find difficulty in following the author of the article; many sentences may necessitate repeated reading—or we might even say "studying"—but he will find it worthwhile doing so, whether he happens to be directly engaged in industrial management or subject to its guidance, that is to say, an industrial employee. We say this because his understanding of the real meaning of management will be vastly clarified, with consequently greater benefit to progress.

* * *

In connection with the publication of the article entitled "Design and Industrial Applications of the Shunt-Commutator Motor," in THE ENGINEERS' DIGEST, October and November, 1948, pp. 330-334 and pp. 379-382 respectively, the author of this article, M. Melone, Chief Engineer of Tecnomasio Italiano Brown Boveri, has asked us to emphasise the fact that this article deals with developments and investigations carried out by Messrs. Brown Boveri & Co., Ltd., of Baden, Switzerland, and that in particular the "Multiplex" winding is based on patents of this firm.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

ALUMINIUM

The Oxygen Efficiency in Anodic Oxidation of Aluminium

By J. KRONSBEIN. (From *Journal of The Electrochemical Society, U.S.A.*, Vol. 94, No. 6, December, 1948, pp. 353-366, 11 illustrations.)

THIS paper describes experiments which lead to the abandonment of the view that anodic coatings on aluminium " grow inward " because of slow chemical solubility of the oxide. It introduces the idea of " oxygen efficiency," which represents the fraction of electricity passed through an anodizing cell which is responsible for the formation of aluminium oxide, the remainder causing either generation of free oxygen or anodic dissolution of the metal. This leads to simple algebraic expressions permitting correlation of current density, temperature of electrolyte, and concentration with this efficiency. Graphs of measurements of this nature indicate a close parallelism with an abrasion test used for evaluating the quality of anodic coatings. Subsequent adjustment of the theory to take into consideration chemical solubility of oxide leads to mathematical expressions showing that the oxide coating grows to a maximum thickness, in agreement with experimental results in an important paper by Mason and Slunder.* An expression for oxide loss by chemical solution is also derived which is supported by measurements in the same paper.

*" Anodic Reactions of Aluminium and its Alloys in Sulphuric and Oxalic Acid Electrolytes."—*Ind. Eng. Chem.*, **39**, 1602 (1947).

METALLURGICAL ENGINEERING

Development of Cleavage Fractures in Mild Steel

By A. B. BAGSAR. (From *ASME Transactions*, Vol. 70, No. 7, October, 1948, pp. 751-809, 50 illustrations.)

THE susceptibility of several types and thicknesses of mild steel of ship-plate and pressure-vessel qualities and of samples of welds to development of cleavage or brittle fractures has been determined by a new test, termed the " cleavage-tear " test, in which a notched tensile-bend type of test piece is used. The effects of notch and test-piece geometries, load concentricity, rate of loading, testing temperature, and of heat treatments were investigated.

At temperatures below the transition-temperature range, the presence of a notch of proper geometry and orientation was found to create a state of multi-axial stress which appears to expand the elastic range of steel and to raise its proportional limit almost to coincide with its breaking point. The breaking strength of rectangular sections containing a notch of the foregoing type was found to decrease, and susceptibility to cleavage fracturing to increase as the notch root radius was decreased. Within the limits investigated, the breaking strength of steel was found to be modified by the notch geometry, this modification being practically independent of the size effect if the section was 6 in. or greater in depth. The transition temperatures for the steels investigated in the as-rolled condition were found to be about 100° F higher than those indicated by the conventional Charpy impact test, and are considered to be more nearly indicative of the behaviour of steels in service. Nevertheless, the transition temperature of a steel cannot be defined for service conditions unless the state of stress which prevails under the same conditions is also defined.

Two types of fracture were encountered, one being the brittle or cleavage type exhibiting no appreciable deformation and lower nominal breaking strength, and the second the shear type exhibiting normal ductility and higher breaking strength. Fractures above the transition range were of the shear type and below that temperature range of the cleavage type. On the basis of the test data presented, reconsideration of our present factors of safety is suggested for monolythic structures with the object of safeguarding against the possibility of development of large cleavage fractures from small discontinuities, notches, or cracks. Other remedial measures for minimizing the damage of cleavage fractures, including modifications in design and material of construction, are suggested.

METALS CORROSION

On the Corrosive Attack of Motor Fuels

By VON P. SCHLÄPFER and A. BUKOWIECKI. (From *Schweizer Archiv, Switzerland*, Vol. 14, No. 9, September, 1948, pp. 257-274, 10 illustrations.)

AFTER discussing the available literature on the corrosive attack caused by motor fuels, and after dealing with some physico-chemical considerations, some of the authors' tests are described. Chiefly, the corrosion-promoting effect of acid and water contained in the motor fuels was examined. The investigations were carried out with typical motor fuel components and blends, using different metals; exposure was usually for one week at 30° C.

It has been shown that in motor fuels free from water, small quantities of acids generally play a subordinate part with regard to the corrosive behaviour. Only in alcohols do they promote corrosion, the intensity of which, for the same addition of acid, diminishes with increasing molecular weight of the alcohols.

Serious corrosion appears in the case of various motor fuels containing water, as, under certain conditions, the water can separate on the metal surface as a result of which conditions for electro-chemical corrosion arise. The tendency to this kind of attack increases at the same water content with decreasing polarity of the motor fuel; e.g., with alcohols of lower polarity, this tendency is weak, but on the contrary, it is very strong with motor fuel blends. Pure hydrocarbon fuels containing very little dissolved water can only markedly attack metals when they are in contact with water layers or drops.

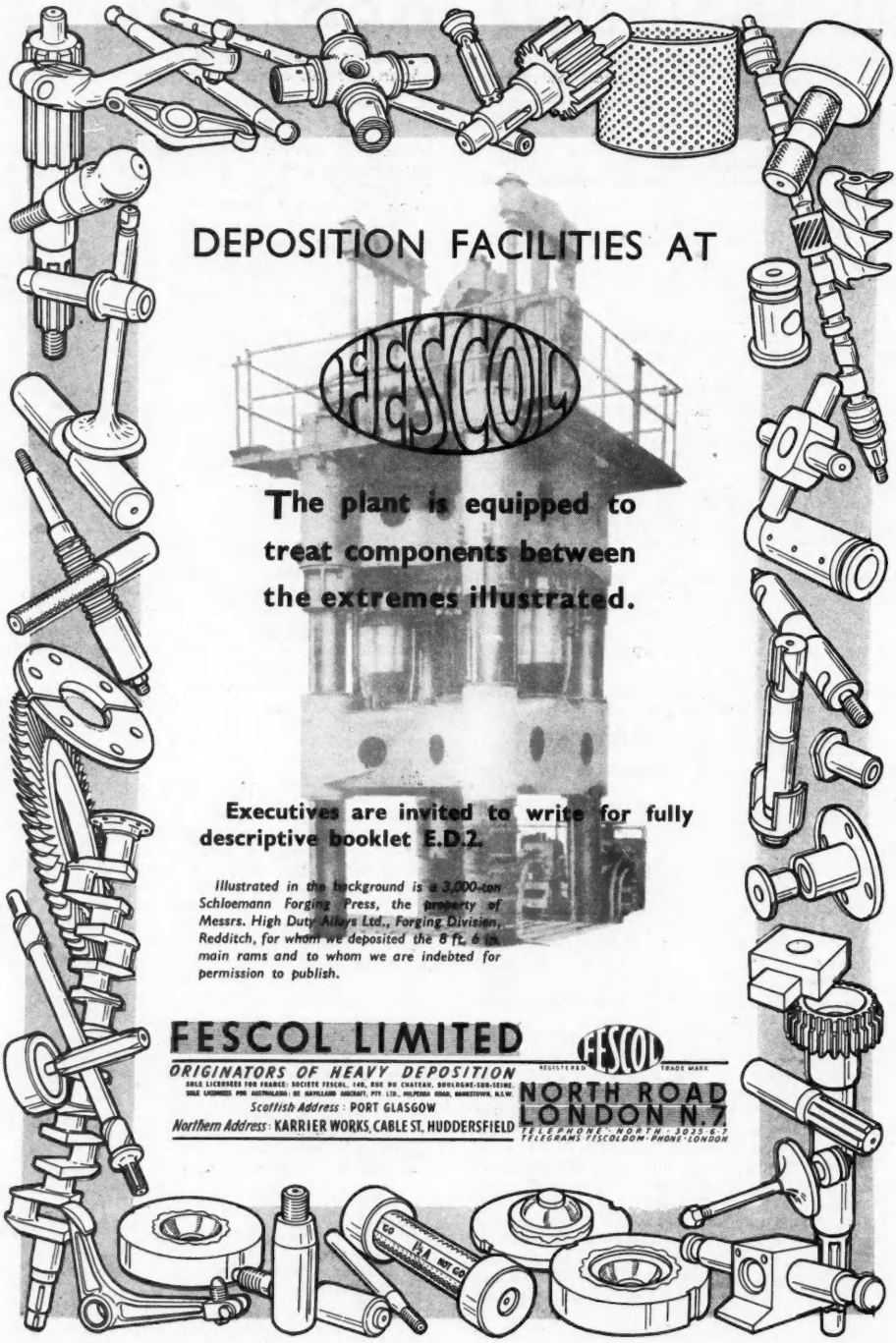
Based on the results of this investigation, the motor fuels have been classified in definite groups with regard to their corrosive behaviour.

OXYGEN

The Planning of Large-Scale Oxygen Production

By E. KARWAT. (From *Stahl und Eisen, Germany*, Vol. 68, No. 25/26, December 2, 1948, pp. 453-465, 15 illustrations.)

OXYGEN is of industrial interest in the production of steel, gas and synthetics, and its use has become widespread since the introduction of the Linde-Fraenkl



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Oxygen—*continued.*

system of fractional distillation of liquid air in 1928. With this system, the production of really pure oxygen is more expensive than that of, say, 75 per cent oxygen, and in planning a set of equipment the most economical purity has to be decided. In the synthesis of ammonia and benzine, in view of the cumulative difficulties encountered in the hydrolysis stage of the cycles, purities of at least 98 per cent are essential. However, where oxygen is used in connection with a combustion process, as in the iron and steel industry, high purity is not required, and economic considerations predominate. These are complicated, as more oxygen is required if its purity is low, until it is realized that the utility is determined by the elimination of nitrogen from the heating cycle, which saves the heat imparted to the nitrogen and increases thermal efficiency. The elimination of nitrogen then becomes the quantitative aim of oxygen production, and costs and production requirements should be reduced to these units. Theory predicts a steady decrease in energy requirements with decreasing purity of the product, and practical data corroborating this conclusion are available. Costs are also influenced by the size of the plant. This is especially so with the smaller sizes, but it is thought that beyond an air consumption of about 20,000 Nm^3/hr, the cost of the equipment is governed by the heat exchangers of the area in the separator, so that it is roughly proportional to the output. It is argued, especially in the United States, that much larger plants would be economical in view of their simplicity, but this is doubted here, as reserve capacity for periods of cleaning and maintenance is often required, which would be more expensive with larger units. Important components of the equipment are the compressors. To obtain freedom from oil contamination, and on the grounds of economy, turbo-compressors are often used. They are driven either by steam turbines or electrically, the latter being particularly suitable for standby units due to quick starting. The main advantages of steam turbines are ease of regulation, and in many cases, that boiler capacity may be available or that use could be made of waste heat boilers. Economical compression may also be obtained from reciprocating gas-engine-driven compressors, if blast furnace gases are available. Waste gases have also been used to drive expansion turbines directly and detailed data of such designs are available. Choice between these possibilities will be governed by local conditions, the ruling prices of each type of equipment and the overall size of the plant. Cooling water problems are discussed in some detail, and the use of cooling by the expansion of the extracted nitrogen to atmospheric pressure is mentioned. Before arriving at a detailed specification, once the purity and level of output have been settled, an analysis of the components of the total cost is revealing. Unlike most processes, there is no material cost. Energy cost, capital charges, wages and maintenance are compared for alternative designs of an industrial station of about 41,000 Nm^3 nitrogen extraction. While the actual prices are difficult to interpret, owing to the use of pre-war prices for the equipment and astoundingly low prices for electric power, it seems clear that capital charges account for 35-50 per cent, energy costs for 30-50 per cent, and running costs for 10-15 per cent on all designs. It may also be concluded that on the basis of nitrogen extraction, 73 per cent-O_2 production by waste energy costs about 70 per cent of 98 per cent-O_2 produced with steam-driven plant. In connection with the discussion, the overall thermal efficiency of steel-making and refining plant was estimated. It was concluded that not only was production of steel per plant increased, but that in all cases there were increases in the thermal efficiency on the basis of the weight of steel produced.

THERMODYNAMICS

Temperature Measurements in Gases Flowing at High Velocity

By L. MAIMQUIST. (From *Acta Polytechnica, Sweden*, 24, 1948, Mechanical Engineering Series, Vol. 1, No. 4, 52 pages, 26 illustrations.)

IN present-day high-velocity flow processes, as occur in efficiency tests for turbo-compressors and combustion chamber tests for gas turbines or in high-speed wind tunnels, the heat of compression and the heat of friction present special difficulties in temperature measurement. The author's aim is to design temperature probes indicating the temperature of the gas in the state of stagnation after an adiabatic temperature rise. The temperature is measured by thermocouples or the like, but heat losses from the temperature probe affect the accuracy of the indication. A temperature probe developed by Pratt & Whitney Aircraft served as a starting point for the design of an improved probe. The Pratt & Whitney probe, Fig. 1, has been tested and

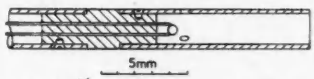

Fig. 1.

showed considerable scatter in the recovery factor $r = (T_i - T_1)/(T_s - T_1)$ where T_i = temperature indicated by thermocouple junction, T_1 = true temperature of gas in undisturbed flow, T_s = stagnation temperature. The instrument finally evolved is shown in Fig. 2, and gives practically complete temperature recovery with $r = 0{\cdot}998$ ($r = 1$ means that the stagnation temperature is measured accurately) or more.

Calculations for the recovery factor and optimum vent hole area are in close agreement with test results.

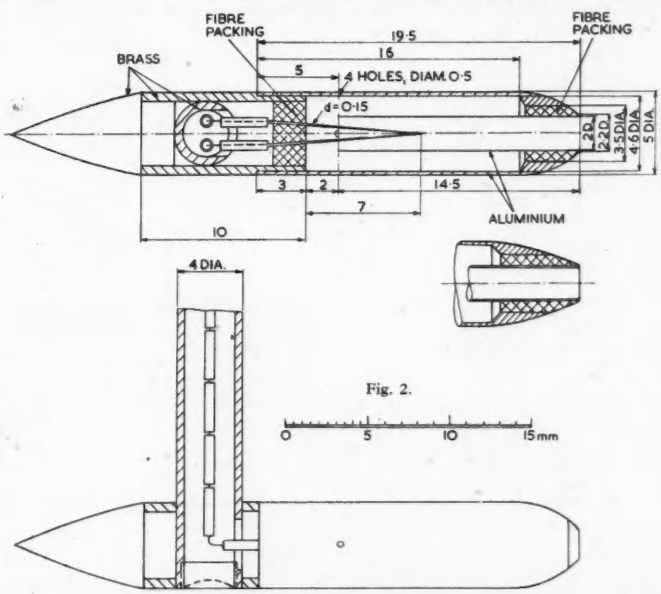

Fig. 2.

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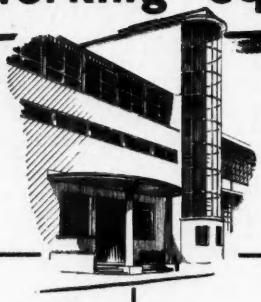

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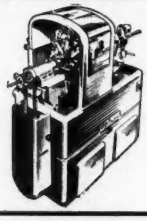

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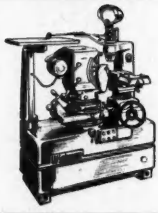

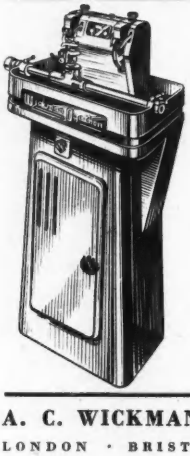

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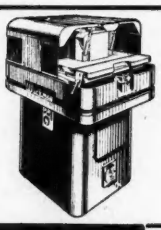

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NEWS OF THE MONTH

PERSONAL

Mr. John Bailey, sales manager and chief technical adviser of Brook Motors Ltd., Huddersfield, has been elected a director of the Company.

Mr. W. W. Balding, head of the experimental department at Cowley, Oxford, has been appointed experimental engineer, **Mr. S. Westby,** standards engineer, and **Mr. H. Rayner,** controller of the Morris and Wolseley chassis drawing office staff of the Nuffield Organisation.

Mr. Frank A. Benson, M.Eng.(L'pool), A.M.I.E.E., has been appointed lecturer in electrical engineering at the University of Sheffield.

Mr. P. R. Betts has been appointed director of A. H. Hunt Ltd., Bendon Valley, London, S.W.18.

Mr. J. Blair, assistant mechanical and electrical engineer, Scottish Region, British Railways, has been appointed carriage and wagon engineer, London Midland Region, Derby.

Mr. L. H. Bolton, Sir George R. Campbell, K.C.I.E., Mr. G. Hinton, Mr. R. Kelso, Mr. W. E. Keville, C.B.E., Mr. E. A. Lloyd, Mr. W. C. Warwick, A.Inst.N.A., Mr. P. Wharton, Captain Sir I. Hamilton Benn, Bt., C.B., D.S.O., Sir Charles Davis, Bt., D.L., J.P., and Sir Ralph Metcalfe have been elected to be members of the Port of London Authority for a term of three years.

Mr. E. Brown and Mr. G. A. Phipps have been elected directors of Guest, Keen & Nettlefolds (South Wales) Ltd.

Dr. D. G. Christopherson, B.S., S.M.(Harvard), A.M.I.C.E., Fellow and Bursar of Magdalene College, Cambridge, has been appointed Professor of Mechanical Engineering at Leeds University.

Mr. W. J. A. Davies has been appointed Director of Merchant Shipbuilding and Repairs in the Admiralty.

Mr. E. C. Deering, general manager of the company's Brimsdown works, has been appointed joint managing director of Johnson, Matthey & Co., Ltd., Hatton Garden, London, E.C.1.

Mr. C. R. Elliott, B.Sc., A.M.I.E.E., has been taken as a partner into the firm of Messrs. Coode, Vaughan-Lee, Frank & Gwyther, 9 Victoria Street, London, S.W.1.

Mr. Edgar Fielden has been appointed director of W. H. Dean & Son, Ltd., Burnley.

Mr. J. P. Ford has been appointed joint general manager of British Oil Engines (Export) Ltd., 32 Duke Street, London, S.W.1.

Mr. H. M. Griffiths, works director, **Mr. D. U. Foster,** secretary, and **Mr. B. Swift** have been appointed directors of Padley & Venables Ltd., Sheffield.

Mr. A. J. K. Honeyman, B.Sc.(Glas.), A.R.T.C., F.I.M., has been appointed chief metallurgist of the Steel Company of Wales Ltd.

Dr. S. G. Hooker, O.B.E., B.Sc., D.I.C., A.F.R.Ae.S., has joined the Bristol Aeroplane Co., Ltd., as assistant to Mr. F. M. Owner, chief engineer of the Aero-Engine Division.

Wing-Commander N. J. Hulbert, M.P., has been elected a director of Metropolitan-Vickers Electrical Export Co., Ltd., Trafford Park, Manchester 17.

Mr. Philip R. Mossay, M.A., A.M.I.Mech.E., has been elected director of Laurence, Scott & Electromotors Ltd., Manfield House, Southampton Street, London, W.C.2.

Mr. Mark Norman has been appointed director of the Staveley Coal & Iron Co., Ltd.

Mr. C. Ian Orr-Ewing, A.M.I.E.E., has been appointed director of Cossor Radar Ltd., Highbury Grove, London, N.5.

Mr. H. B. Robin Rowell, C.B.E., A.F.C., D.L., M.I.N.A., chairman of R. and W. Hawthorn, Leslie & Co., Ltd., has been re-elected president, and **Mr. Henry Main, C.B.E., J.P., M.I.N.A.,** managing director of the Caledon Shipbuilding and Engineering Co., Ltd., vice-president of the Shipbuilding Conference.

Mr. H. L. Satchell, M.B.E., manager of the British Thomson-Houston Co., Ltd., Rugby Works Group, has been appointed a director of the Company.

Mr. Hugh C. Tett, B.Sc., A.R.C.Sc., D.I.C., M.Inst.Pet., managing director of the Esso Development Co., Ltd., has been appointed general sales manager of the Anglo-American Oil Co., Ltd., 36 Queen Anne's Gate, London, S.W.1.

Mr. P. G. Walker has been appointed director of Ambrose Shardlow & Co., Ltd., Sheffield.

Colonel R. J. Walker, O.B.E., R.E., has been appointed inspecting officer of railways in the Ministry of Transport.

Mr. Gordon Weston, B.Sc. (Eng.) has been appointed assistant director of the British Standards Institution, 24-28 Victoria Street, London, S.W.1.

Mr. H. W. Winham has been appointed manager, **Mr. H. H. E. George,** sales manager and **Mr. W. E. J. Adams** continues as works manager of Armstrong Whitworth & Co. (Pneumatic Tools) Ltd., Close Works, Gateshead-on-Tyne.

Mr. W. B. Yeadon, A.M.I.Mech.E., has been appointed home sales manager of J. H. Fenner & Co., Ltd.

Admiralty Ship Welding Committee. The Second Interim Report of the Committee is to be published shortly by His Majesty's Stationery Office. This report reviews the work of the Committee since the First Interim Report, and covers :—

(i) Collection and analysis of information on casualties to welded ships.
(ii) Design, and its relation to failures, with particular reference to the differences in structural behaviour between riveted and welded ships.
(iii) Workmanship covering not only welding technique, but also methods of construction and their relation to the behaviour of welded ship structures.
(iv) Material, with particular reference to the suitability of ship quality mild steel for the construction of all-welded ships.

The frequency of major failures in welded ships has diminished considerably over the past few years, and this may be attributed to improvements resulting from the work of the American and British Committees.

Important full-scale investigations on ship structures carried out by the Committee includes :—

(1) Hogging and sagging experiments on the all-welded tanker *Neverita*.
(2) Hogging and sagging experiments on the similar riveted tanker *Newcombia*.
(3) Seagoing trials on a welded cargo ship s.s. *Ocean Vulcan*. This ship completed eight round voyages across the North Atlantic over a period of 17 months, during which a very large number of observations were made, including the distribution of water pressures on the hull, accelerations, stresses, wind forces, roll and pitch, etc.
(4) Hogging and sagging experiments on the s.s. *Ocean Vulcan*.
(5) Hogging and sagging experiments on the similar riveted cargo ship *Clan Alpine*.

These experiments have been on a greater and more thorough scale than any previous similar investigations and have been directed towards obtaining direct comparisons between similar welded and riveted ships.

The Report gives full details of each aspect of the Committee's work including a list of fifteen references to relevant research reports and memoranda, many of which have been issued by the Committee itself.

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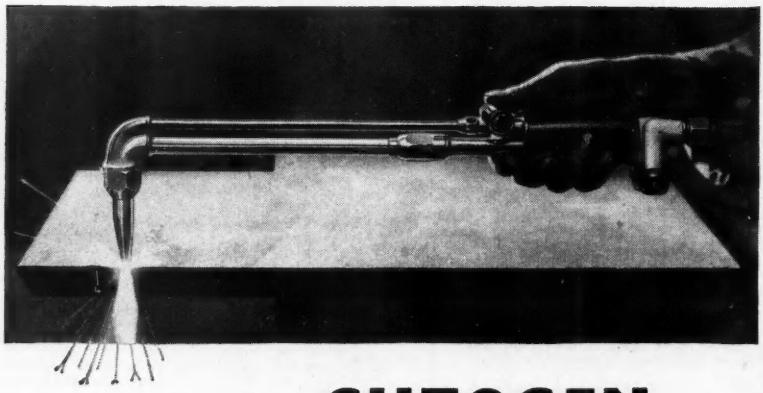

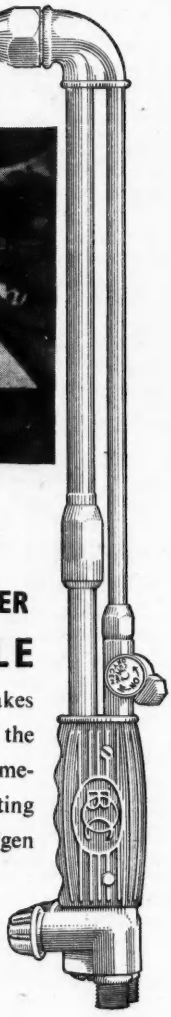

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The summary and conclusions of the Second Interim report deal among other subjects with continued investigations on such important questions as design, locked up stresses, workmanship, modern methods of inspection, including radiography, the quality of materials and the notch brittleness of certain mild steels.

"Measurement of Productivity." The Joint Committee of the Institution of Production Engineers and the Institute of Cost and Works Accountants, which is investigating means of measuring productivity and devising methods of applying this measure for increasing production, has met four times since the One-Day Conference held in London in November.

Well known manufacturing organisations in London are co-operating in the work of the Committee, and Trade and Technical Associations are showing a keen interest in the subject.

American Management Techniques. Mr. A. P. Young, O.B.E., Chairman of the Institution of Works Managers, said at a luncheon meeting of the Institution that: "The average level of productivity in the U.S.A., is now a little more than twice as great as in Britain. Fundamentally, because the horse-power harnessed to every American worker, it is correspondingly greater. If we compare 'individual productivity,' there is nothing in it. In my study of two mass-production factories—one in Boston and the other in London—which were strictly comparable as regards product, lay-out and manufacturing methods, I found that *productivity was identical*. The problem facing British industry is therefore to raise the horse-power at the command of the worker."

"The prelude to this is simplification and standardisation of design, before the manufacturing lay-out is planned. As has been well proved in the United States, raising productivity is largely a question of eliminating waste in three main directions: mechanical and heat energy; labour, by hand and brain; and materials."

"In the United States, the status and authority of the works manager is higher than here, and he is given full control of all the productive functions. British Works Management can learn much from American techniques and practices."

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"The 'housekeeping' in American factories is greatly superior to our own. A National Clean-Up Week for the whole of industry might be launched by Sir Stafford Cripps, and the scrap recovered would surprise everyone."

"American industry is strikingly ahead of our own in mechanical handling. This helps to explain the higher ratio between direct and indirect workers which prevails in the U.S.A. A study of layouts will generally reveal how the 'flow-line' can be straightened out by moving plant and equipment."

"Suggestion schemes are widely adopted throughout American industry, and generous rewards are paid."

"The successful works manager must function largely as a teacher and, like all good teachers, he must acquire skill in drawing the best out of his team."

BOOKS

Engineering Tolerances. A study of tolerances, limits and fits for engineering purposes, with full tables of all recognized and published tolerance systems. By H. G. CONWAY, M.A., A.M.I. Mech.E., F.R.Ae.S. Publishers: Sir Isaac Pitman & Sons, Ltd. Parker Street, Kingsway, London, W.C.2. 286 pp., 103 diagrams and tables. PRICE 30/-.

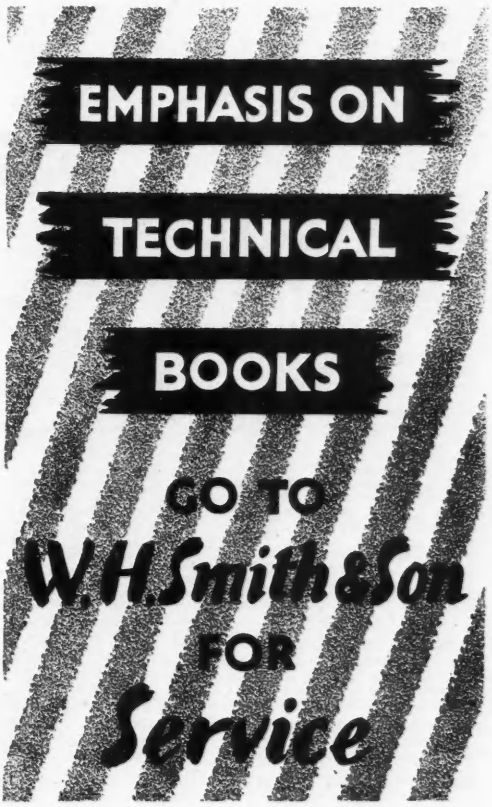
The author says in the preface *inter alia*: "This book is an attempt at a clarification of the position, the prime object being to persuade the designer to use tolerances and limits with sincerity and genuine intention, both shop man and inspector then being in a position to treat the designer's drawings with more respect than is current in most production shops to-day."

The author has to be congratulated not only to this intention, but also to the way, how he has succeeded in his task to make clear his difficult subject in such a comprehensive manner. There is, in our opinion, no English publication equal to this book, and the often quoted phrase is certainly true in this case, that it fills a gap which has been felt by many, who are really interested in the future of the British engineering industry.

F.B.I. Register of British Manufacturers—1948/9. 21st Edition. Published for The Federation of British Industries, by Kelly's Directories Ltd. and Iliffe & Sons, Ltd. 752 pages. PRICE 42/-.

The 1948/9 F.B.I. REGISTER lists nearly six thousand firms and their products, and includes much information of interest to overseas buyers. Especially valuable is the Products & Services Section, with more than 5,000 product headings facilitating the rapid identification of supply sources, the Address Section giving complete information on every Member, and the Brands & Trade Marks Sections indicating the makers of branded products. There are reference facilities in French and Spanish, while the classification has been improved and greatly extended.

The new F.B.I. REGISTER will have world-wide distribution to importers, buyers, trade commissioners, chambers of commerce, etc., and will provide valuable support for the export drive. Sir Frederick Bain, M.C., President of the F.B.I. and Deputy Chairman of I.C.I., says in his foreword: "We believe this Register is a real service . . . in these days when, owing to the scarcity of materials, orders have to be spread over many different firms, the Register becomes an increasingly valuable sales instrument."



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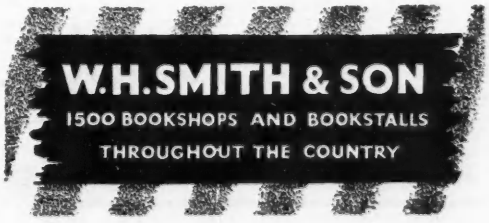

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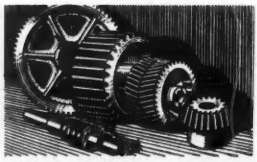

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Elementary Mechanical Vibrations. By AUSTIN H. CHURCH Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. 200 pp. PRICE 22/6.

This book is based on courses given by Professor Church to various groups of students at New York University. It can be regarded as an extremely useful introduction into the complex subject and a stimulant to further and more advanced study of vibration problems. Self-excited vibrations, Non-linear systems, Harmonic analysis, and Damping in systems having more than one degree of freedom, are not dealt with in this book as they were regarded beyond the scope of the text. A chapter covering the basic principles of Balancing, a subject so intimately related to Vibration, is included, however.

Reviews of Petroleum Technology, Vol. 7. Publishers: The Institute of Petroleum, 26 Portland Place, London, W.1. 535 pp. PRICE 21/-.

This volume collects, collates, and reviews references to over 3,000 articles which appeared during 1941 to 1945. It is a much needed and valuable guide to available data on the subject which is covered in these Reviews under the following headings:— Petroleum Geology; Geophysics; Drilling; Production Engineering; Petroleum Transportation in the U.S.A.; Refinery Process Developments; Analysis and Testing; Chemistry of Petroleum; Physics of Petroleum; Natural Gas, Liquefied Petroleum Gases and Natural Gasolene; Benzole; Alternative Fuels: Fuels Produced by Hydrogenation and Synthetic Processes and Low and Medium Temperature Carbonization and other Substitute Fuels; Specifications for Service Fuels; Diesel Fuel and Gas Oils; Wartime Specifications for Lubricants; Asphaltic Bitumen and Road Materials; Special Products; Petroleum Literature.

The Welding Review Year Book, 1949. General Editor: J. V. Brittain. Publishers: Engineering Trader Ltd., 26-30 Holborn Viaduct, London, E.C.1. PRICE 8/6.

Friends of the earlier edition of this little book will be pleased to see after war and post-war troubles a new edition with a completely new text, brought up-to-date and comprehensive, as far as possible in the limited size of the book. Of the some 30 chapters, only the following may be mentioned, in order to give the reader an impression of the variety of subjects this publication is dealing with: Institute of Welding; Welding training; Arc welding; Stud welding; High frequency welding; Thermit welding; Butt, Spot, Seam, Projection welding; Brazing; Oxy-acetylene welding; X-rays in welding; and especially the 40 pages of weldability tables.

Year Book of the Heating and Ventilating Industry. Publishers: Technitrade Journals Ltd., 8 Southampton Row, London, W.C.1. 206 pp. PRICE 5/- net.

In addition to the Membership List of the Association of Heating, Ventilating and Domestic Engineering Employers and other information concerning the Association, the Yearbook contains a number of illustrated articles on subjects of interest to the profession, such as Panel Heating, by H. H. Bruce; Fuel Economy as it affects Heating Installations, by J. O. Cooke; Fuel Oil Conversion Schemes, by F. J. Erroll; Types and Scope of Ventilating and Air-Conditioning Installation, by Leslie Rayner; Technical Education, by Fredk. J. Packer; other articles deal with Temperature Measurement and Comfort; Ventilating Design; and other subjects of interest especially to contractors.

The Instrument Manual, 1949. Publishers: United Trade Press Ltd., 24 Bride Lane, London, E.C.4. 548 pp. PRICE 70/-.

The publishers with the co-operation of the authors, D. Brownlie, H. B. Cronshaw, A. E. Crawford, R. E. Iggleden, L. B. Lambert, A. Linford, J. T. Miller, A. Muir, E. Rourke, P. Strip and L. Walter, and with the assistance of numerous firms of the British instrument industry, have succeeded in producing a well-edited and valuable collection of information of much interest to all concerned with industrial instruments.

The book gives detailed description of a wide range of instruments; data which are frequently needed by users of industrial instruments; a valuable bibliography; and a directory of manufacturers and organisations directly interested in the subject.

Angewandte Hydraulik. By PROFESSOR ROBERT DUBS. Publishers: Rascher Verlag, Zurich. 405 pp., 180 Figs.

There are many good and recent books on Applied Hydraulics, written for the practical engineer in the English language. The number of German books on the subject appears to be equally large. The author of the present work, a professor of the Swiss Technical College, felt, however, that there is need for a comprehensive book dealing with a great variety of problems encountered by the practising engineer, in which, as far as possible, all problems should be dealt with independently, and the mathematical relationship is presented in full derivation. Moreover, theoretical derivations are confronted with experimental results obtained in the hydraulics laboratories of the Federal Technical College. These are published for the first time.

The book, presenting the lates. results of research and dealing with all problems from hydraulics to non-stationary flow in a clear manner, can be thoroughly recommended. Excellent figures, examples and a long list of bibliography adorn this useful work.

Arbeitsverfahren und Betriebsmittel. By OSKAR GÖNNER. Publishers: Carl Hanser Verlag, München. 176 pp., 244 Figs. PRICE 8.20DM.

This book deals with a great number of methods for the manufacture of pipes and tubular hollow parts, and it represents a commendable effort in imparting knowledge in this rather neglected field. In point of fact, its special merit lies in drawing attention to obsolete practices and in giving full consideration to modern methods.

In dealing with methods of manufacture, emphasis is laid on thin-walled tubes (0.5 to 2 mm.) such as generally used in pipelines for the conveyance of fuel, water, lubricants, compressed air, etc. Apart from a few exceptions, the upper limit for the diameter of such tubes is taken to be about 80 mm. This applies to semi-finished products as delivered by the rolling mills. The various types of conical tubes, bends, distributors and other complicated shapes manufactured to any size by fabrication from sheet metal are also fully described in the present book. Numerous excellent illustrations, many of which give dimensions, accompany the text.

Turret Tooling. By H. HOWARD FREEMAN. Second Edition. Publishers: Sir Isaac Pitman and Sons, Ltd., Parker Street, Kingsway, London, W.C.2. 136 pages. PRICE 12/6.

The author has succeeded in his aim "to be of assistance to all who are responsible for estimating and planning production and to those who actually use and set the tools in the machine shop."

Whilst the introductory section deals with tool selection and examples of tool lay-out, the two main sections are devoted to Capstan and Turret Lathe Tools and to Turret Lathe Attachments. The numerous appendices may give to a large extent figures which are available elsewhere; they are, however, not only very handy in this book, which is to a large extent in the hands of real "shop-men," but also valuable because they draw the attention to British Standards which are not sufficiently used in actual practice.

Production Engineering. Practical Methods of Production Planning and Control. By T. S. MURPHY. Publishers: The Louis Cassier Co., Ltd., Dorset House, Stamford Street, London, S.E.1. 226 pages.

This book is concerned with basic principles which can and should be adapted to local condition as required. It is intended for students of production engineering and for young planning engineers.

In ten chapters it deals with: (1) the work and the organization of the design and drawing office; (2) with workshop and process layout, especially jig, tool and fixtures' design and manufacture; and (3) with production control (a) from the administrative point of view; (b) as far as time and motion studies are applied; and (c) the principles of estimating and costing are to be taken into consideration.

The individual chapters include a list of more specialised books which are recommended for more detailed studies.

Illustrated Jig-Tooling Dictionary. By T. G. THOMPSON and R. A. PETERSON. Publishers: Macmillan & Co., Ltd., St. Martin's Street, London, W.C.2. 349 pp. PRICE 37/6 net.

This is an excellent book of the greatest practical value to all interested or actively engaged in jig construction and tool engineering. Over 1000 terms are briefly explained with admirable clarity. Each term is illustrated by a functional drawing which is beyond criticism. All data have been taken from the design and construction of accual jigs, tools and fixtures in use in modern industry.

Turning and Boring Practice. By FRED. H. COLVIN and FRANK A. STANLEY. Publishers: McGraw-Hill Publishing Co., Ltd., Aldwych House, London, W.C.2. 530 pp. 76 Figures. PRICE 28/6.

This is the third edition of this well-known comprehensive book dealing with turning and boring practice. Latest data on mandrel and taper work in lathes, on precision boring of extreme accuracy, on boring bars for special work and on carbide tools, have been incorporated in this new edition to bring the book up-to-date.

Organisation and Control of Production. This Paper which explains the system of production control of the Associated Equipment Co., Ltd., was first published in the December, 1943, issue of the Journal of the Institution of Production Engineers. It has now been reprinted for the second time and it is available at 5/- per copy.

BUSINESS NOTES

Service for Diesel Engines. Some 250,000 Diesel engines were produced by General Motors, the majority of which were supplied to the armed forces of Great Britain and the U.S.A., during the war.

Of this enormous number a very large proportion have found their way into the hands of private users in this country, where they are still performing yeoman service in assisting Britain's industrial programme.

These engines, however, are dependent for efficient performance on really expert servicing and the use, for replacement purposes, of genuine factory spare parts.

In addition, there were certain types of wartime engines which were given quite abnormal tuning in order to satisfy a wartime requirement. These engines require to be de-tuned if they are to perform a normal peacetime duty in a reliable manner. This de-tuning requires factory instructions and sometimes special replacement parts.

General Motors Limited, of 23 Buckingham Gate, London, S.W.1 (telephone: Victoria 8141) wishes to announce to all owners, operators and potential owners of G.M. Engines that it has made available for all models of G.M. engines, whether they are new, second-hand or "war-surplus," whether converted to other uses or in the original form, its full factory-sponsored service facilities.

The organization of this Service consists firstly of the expert advice and assistance of its factory-trained staff of technicians, who are generally experienced in the wartime use of the engines as well as trained in what their peacetime performance should be, to advise on installation adaption and reconditioning problems. Secondly, on the efficient servicing and overhaul of engines including the provision of the factory approved replacement parts, the correct

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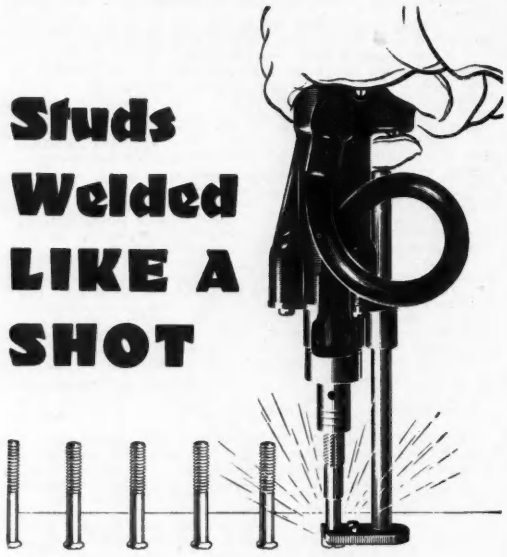

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adjustment and tuning of injectors, and the dynamometer testing to factory-approved performance figures. Thirdly, the supply of genuine factory-approved spare parts, stocked in this country.

A staff of technicians with fully equipped service vans is available to carry out overhauls on site. Alternatively, engines can be forwarded for replacement by exchange with another reconditioned engine. This replacement scheme is also applicable to sets of injectors, blowers, cylinder heads and other major components.

Water Treatment Plant for a Paper Mill Using Very Polluted River Water. The Pulsometer Engineering Co., Ltd., of Reading have installed a unique water purification plant in Lancashire.

The water before treatment is dark brown in colour with a varying pH according to the quantity and type of industrial effluent discharged into it higher up the river. Large patches of dark foam swirl on the water which is full of suspended matter and chemicals in solution, including sewage.

The plant to treat this water has to draw from the river and produce a clear water of Hazen value of 10 to 15. The raw water sometimes has a Hazen value of well over 500. The plant is designed to give a maximum output of approximately 3 million gallons in 24 hours, and if it had been designed for treatment with Alum only (for precipitation purposes) the slow rate of settling would have necessitated six large conical bottomed settling tanks instead of only two. These two conical towers flank a central square building block and a battery of six rapid open-topped gravity type filters. The central block houses a lift. The storage tanks for the diluted chemicals are situated on the top floor. Water is pumped from the river to a dosing channel in the central block which feeds the two conical towers. Here it is chemically dosed and passed into the concentric-reaction chambers, where gentle agitation takes place. The resultant floc falls to the base of the cones where it can be drawn off periodically as a sludge. The clarified water overflows into a sill on the outer annular portion of the towers and is fed by gravity to the filters.

The principle upon which this clarification normally depends is as follows :—The colloidal matter in the river, which causes colour, froth and slime, carries a negative charge of electricity. If a solution of alumina ferric is added to the water, it gives rise to colloidal particles of aluminium hydroxide which is positively charged. These positive particles will therefore attract the negatively charged impurities in the water. The result is electrically neutral particles which coalesce to form the Alum Floc mass. Alum has been used in the treatment of water in this manner for many years. The Alum floc however is a light one and settles too slowly for efficient use in a continuous conical tower process such as the one put up. It is therefore necessary to produce a heavier floc and use is made of the fact that under suitable conditions, solutions of other chemicals can be treated, so as to produce a colloidal form carrying a negative charge. These particles attach themselves in addition to the organic impurities, to the positively charged Alum. They take no part in the actual process of clarification but add considerably to the weight of the floc and consequently speed up the rate of settling.

Brookhirst Switchgear Ltd., Northgate Works, Chester, announce that as from March 14th, the address of the London office is Alliance House, 12 Caxton Street, London, S.W.1. Tel. WHITEHALL 9904.

Elliot Brothers (London), Ltd., announces a reorganisation of its home sales and service arrangements as follows : London area, Century Works, Lewisham, S.E.13 (telephone, Tideway 3232) ; South-Western area, 37, Mount Stuart Square, Cardiff (telephone, Cardiff 4334) ; Midlands, 9, Caroline Street, Birmingham, 3 (telephone, Central 2037) ; North Midlands and N.W. England, 270-272, Corn Exchange Buildings, Hanging Ditch, Manchester, 4 (telephone, Blackfriars 7795) ; N.E. Coast, Grey's Buildings, 53, Grey Street, Newcastle-upon-Tyne (telephone, Newcastle 22655) ; Scotland and N. Ireland, 55-63, Washington Street, Glasgow, C.3 (telephone, Central 7965).

Everett, Edgcumbe and Co., Ltd., of Colindale Works, Hendon, London, N.W.9, announces the opening of a branch factory at Houghton-le-Spring, Co. Durham. This new manufacturing unit will be concerned with miniature synchronous motors, process timing devices, synchronous clocks and chart drives for roll chart and disc chart graphic instruments. Later other products will be made. The new factory will be under the management of Mr. C. Leney, who was previously in charge of the Instrument Shop at Colindale.

Head, Wrightson& Co., Ltd., of Thornaby-on-Tees, announce that they have acquired the business of F. J. & L. Dean, Slough, Bucks. ; Aldean Designs Ltd., Slough, Bucks. ; and D. J. Hawkins & Sons, Ltd., Stoke Row, Henley-on-Thames, Oxon.

These businesses are being amalgamated with their subsidiary Head Wrightson Light Alloy Structures Limited, designers and builders of the first aluminium alloy bascule bridge in the world erected at Sunderland. The Head Wrightson Light Alloy Structure Company will, in future, be known as Head Wrightson Aldean Limited, with headquarters at 61, Windsor Road, Slough, Bucks.

Industrial Products (Speco) Ltd., 7 Boston Manor Road, Brentford, Middlesex, a subsidiary company of the Sperry Gyroscope Co., Ltd., has concluded a licence arrangement with Air Products, Inc., Emmaus, Pa., U.S.A., for the manufacture in Great Britain, of a new range of oxygen generating plant.

Manchester Oil Refinery Ltd., Petrochemicals Ltd. and **Petrocarbon Ltd.,** have changed their London office address from Adelaide House, London Bridge, E.C.4, to 170 Piccadilly, London, W.1. Tel. MAYfair 6618 (10 lines).

The Mond Nickel Company, Limited, announce that on 28th March, 1949, they are moving their head offices from Grosvenor House, Park Lane, to Sunderland House, Curzon Street. After that date the address of the head offices of The Mond Nickel Co., Ltd., Henry Wiggin & Co., Ltd., and Mond Nickel (Retirement System) Trustees, Ltd., will be Sunderland House, Curzon Street, London, W.1. Tel. ; GROSVENOR 4131.

Rubery, Owen & Co., Ltd., Darlaston, have removed their London office to Kent House, Market Place, W.1. Mr. Gordon Sloan has been appointed general sales manager at Darlaston, and Mr. G. E. Montague has been appointed London manager of the Company.

Wellington Tube Works Ltd. has opened a London office at 36 Victoria Street, London, S.W.1. (Tel. ABBEY 1326) which is headquarters of the London representatives, M. J. A. Blackburn, Mr. G. C. Parks and Mr. A. C. Sorrell.

The Paris International Trade Fair will be held from May 21 to June 6. A very considerable part will be devoted to technical industries. Machine tools, machinery, measuring apparatus, welding equipment, etc., alone cover more than 20,000 square yards of hall space, while open-air exhibits of heavy machinery and material, engines, saw-mill gear, etc., absorb a further 25,000 square yards.

Public works and building equipment, lifting apparatus, scaffolding, road-making material, heating equipment, machinery used in the rubber, plastic, refrigeration, food manufacture, packing and presentation industries are to be found at the Foire de Paris, all bearing the latest improvements of the year.

The Electrical Group assembles a very large number of exhibitors, covering the entire electrical industry. Transformers, dynamos, motors, line installations ; insulators ; domestic appliances ; rural equipment : lighting appliances, including fluorescent lighting ; telephones, telespeakers, control and registering apparatus, scientific apparatus, thermostats, car-equipment, etc., will be shown.

More than 13,000 sq. yds. are devoted to the Electrical, Radio and Television Sections.

Classified Advertisements.

The Rate for all classified advertisements is 6d. per word ; in bold print 9d. per word ; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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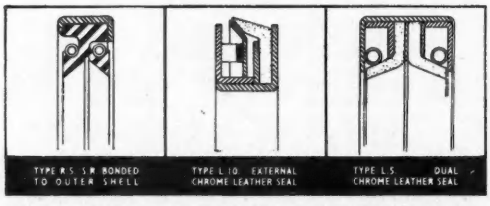

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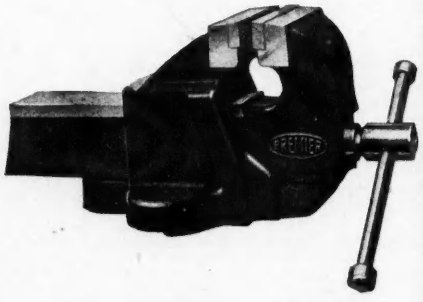

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